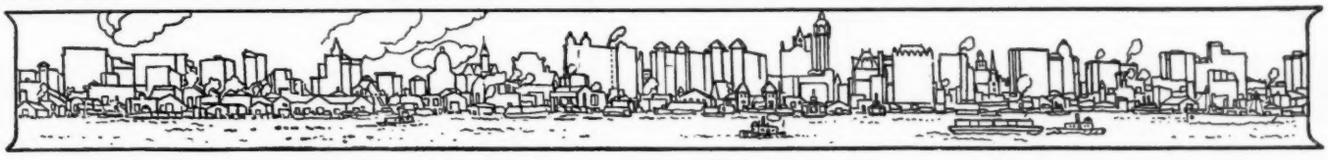
Power. and THE ENGINEER

EVERYTHING ABOUT THE GENERATION AND TRANSMISSION OF POWER



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AT the Philadelphia Centennial Exposition in 1876 the most remarkable feature of the power plant in Machinery Hall was a 1400-horsepower vertical Corliss beam condensing engine, supplied with steam from twenty boilers.

The weight of the engine alone was over 600 tons and the entire plant weighed about 800 tons, or $1\frac{3}{4}$ horsepower per ton.

Some idea of the reduction in the weight of iron necessary per horsepower may be had by comparing the Centennial plant with what is possible today.

Engineers at Schenectady are at work on the design of a steam turbine of 27,000-horsepower capacity, which is twenty times that of the Centennial engine.

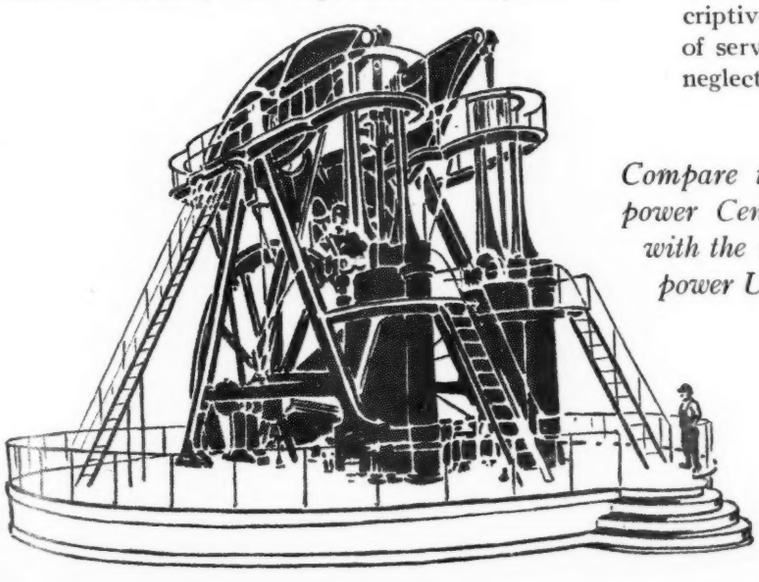
This turbine, with its condenser base and auxiliaries will weigh about 250 tons; an additional weight of approximately 150 tons will cover the generator. Steam will be furnished by fifteen boilers, which, equipped with mechanical stokers, will weigh about seventy-five tons each, making a total weight for the steam plant of approximately 1400 tons. For a 27,000-horsepower plant this would mean nineteen horsepower per ton and a reduction in weight of nearly 90 per cent. over the Centennial engine.

While the weight per horsepower of boilers has been decreased to some extent since 1876, the really long step in weight reduction per unit of capacity has been made with the turbine.

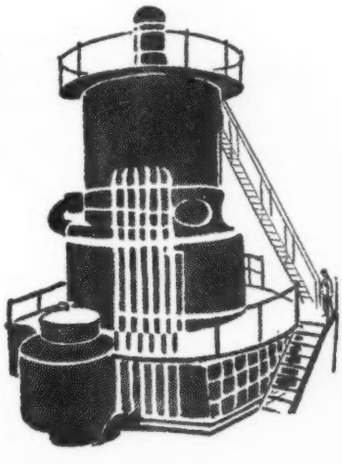
Giving only one-twentieth of the horsepower of the turbine, the Centennial engine weighs three times as much. Thus, while a power-plant equipment of today may weigh but 10 per cent. of what it did 35 years ago, the engine alone need weigh but $1\frac{1}{2}$ per cent.

This foreword is inspired by the fact that this issue contains an exceptional amount of matter upon the steam turbine.

Besides the series of articles on the turbine by Mr. Snow, there will appear, from time to time, others descriptive of turbines of various types and in all classes of service. No engineer who is awake can afford to neglect to inform himself upon this subject.



Compare the 1400-Horsepower Centennial Engine with the 27,000-Horsepower Unit of Today.



Elementary Principles of Steam Turbine

Absolute and Relative Velocities, Fundamental Principles of Commercial Impulse Turbines, and a Comparison with Hydraulic Turbines

BY WILLIAM E. SNOW

ABSOLUTE AND RELATIVE VELOCITIES

In steam-turbine design there are two kinds of velocity, commonly known as *absolute* and *relative*. Absolute velocity may be defined as that velocity which would be absolute if the turbine casing or frame were at rest. The velocity of the earth will therefore be neglected, and

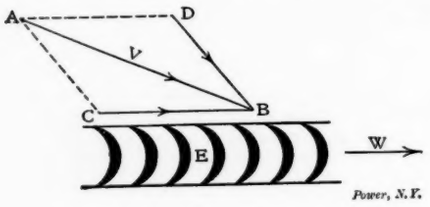


FIG. 1

the velocity of a revolving part of the turbine, or of the operating fluid will be regarded as absolute. Relative velocity is a velocity relative to a moving part of the turbine.

Referring to Fig. 1, assume E to be a turbine wheel, moving with the absolute velocity W in the direction of the arrow. Let V represent the absolute velocity and direction of the impinging jet of steam. Then the velocity of the steam jet, relatively to the turbine, will be obtained by making $CB = W$, and completing the parallelogram $ADBC$, when DB will represent the velocity of the jet relatively to the wheel. This relative velocity DB is the velocity which the jet would have if a velocity were imparted to both the wheel and the jet of an amount sufficient to render the net velocity of the wheel equal to zero. A velocity which would render the net velocity of the wheel equal to zero would be equal in magnitude, and opposite in direction, to W. Therefore, combine this velocity with V, and the velocity DB is obtained.

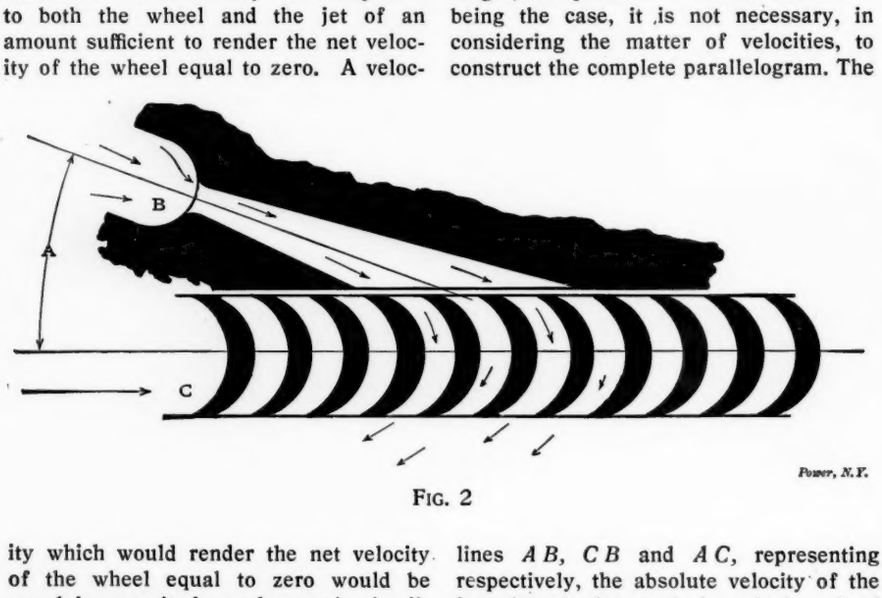


FIG. 2

The velocity of the jet, relatively to the wheel, may also be otherwise defined as that velocity which, combined with the velocity of the wheel, produces the absolute velocity of the jet. In the diagram, DB represents the velocity which, combined with the velocity of the wheel, produces the absolute velocity AB of the jet. Therefore, DB represents the velocity of the jet relatively to the wheel.

As the figure $ADBC$ is a parallelogram, the line AC must be equal in length, and parallel to the line DB. This being the case, it is not necessary, in considering the matter of velocities, to construct the complete parallelogram. The lines AB, CB and AC, representing respectively, the absolute velocity of the jet, the absolute velocity of the wheel and the velocity of the jet relatively to the wheel are all that is required. In the following graphical illustrations of the various types of impulse turbines this method of diagraming the velocities by means of velocity triangles will be followed.

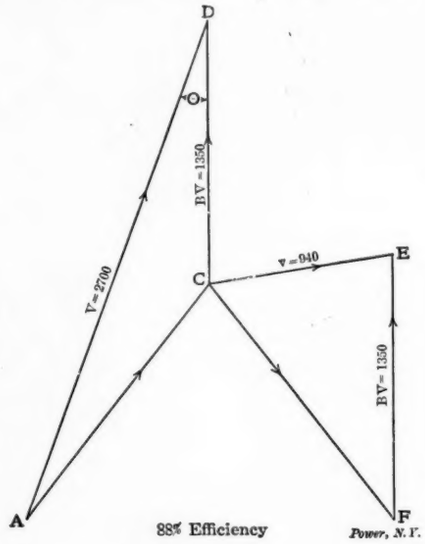


FIG. 3

The absolute velocity of the fluid when inclosed by the vanes is not important, but the absolute velocities when entering and leaving the vanes are, as the kinetic energies of the fluid when entering and leaving the vanes are proportional to the squares of these velocities. Let V be the absolute velocity of the fluid when entering the vanes, and v the absolute velocity of the fluid when leaving the vanes. Then the kinetic energy given up by the fluid to the turbine will be proportional to $V^2 - v^2$, and the efficiency, neglecting friction, will be,

$$\frac{V^2 - v^2}{V^2}.$$

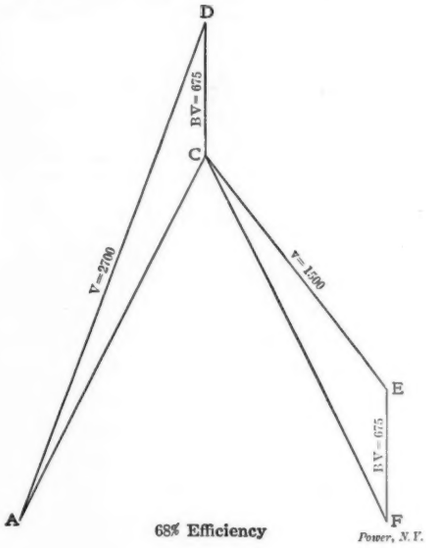


FIG. 4

It will be seen that the angle of the vanes, except at the points of entrance and exit, cannot affect the efficiency, except through increasing or diminishing the frictional losses. A smooth, gradual curve is all that is required for the vane between entrance and exit angles.

TYPES OF IMPULSE TURBINE

The various types of turbine of the impulse class, showing in a graphical manner by means of suitable velocity diagrams their methods of steam utilization, will now be considered. One of the chief problems that presents itself to the designer of a steam turbine of the impulse class is that of the efficient utilization of exceedingly high steam velocities. The velocity of steam in an

expanding nozzle is dependent upon the initial or entrance pressure, and the final or exit pressure (also termed back pressure). The accompanying table gives the velocity of steam in nozzles at varying initial and final pressures. From this table it will be seen that the steam velocities of the single-stage impulse turbine vary all the way from 2587 feet per second in a noncondensing machine operating on a steam pressure of 80 pounds, to 4057 feet per second in a condensing turbine operating on a steam pressure of 160 pounds, with a vacuum of 28 inches.

VELOCITY OF STEAM FROM NOZZLES.

Initial Pressure, Pounds per Sq.In. Gage.	Velocity in Feet per Second at Varying Back Pressure. Back Pressure 1 Atmos. (14.7).	Back Pressure 25 In. Vacuum.	Back Pressure 28 In. Vacuum.
80	2587	3419	3798
90	2648	3458	3849
100	2700	3526	3881
110	2762	3564	3918
120	2815	3605	3953
130	2859	3638	3978
140	2907	3670	4011
150	2963	3692	4036
160	3009	3728	4057

These figures are for dry saturated steam only.

The Single-stage Turbine—In the single-stage turbine, where the steam acts but once on the bucket wheel, the best efficiency is obtained when the peripheral speed of the wheel, or as it will be hereafter termed, bucket velocity, is approximately one-half that of the impinging steam jet. Fig. 2 shows the nozzle and bucket arrangement of a turbine of this type, in which *B* is the nozzle and *C* the bucket wheel. The angle *A* of the nozzle relative to the plane of motion of the buckets varies from 17 to 25 degrees in different types of commercial turbine. In the present case it is shown as 20 degrees, which is the angle most commonly used. The flow of the steam through the nozzle and buckets is shown by the small arrows.

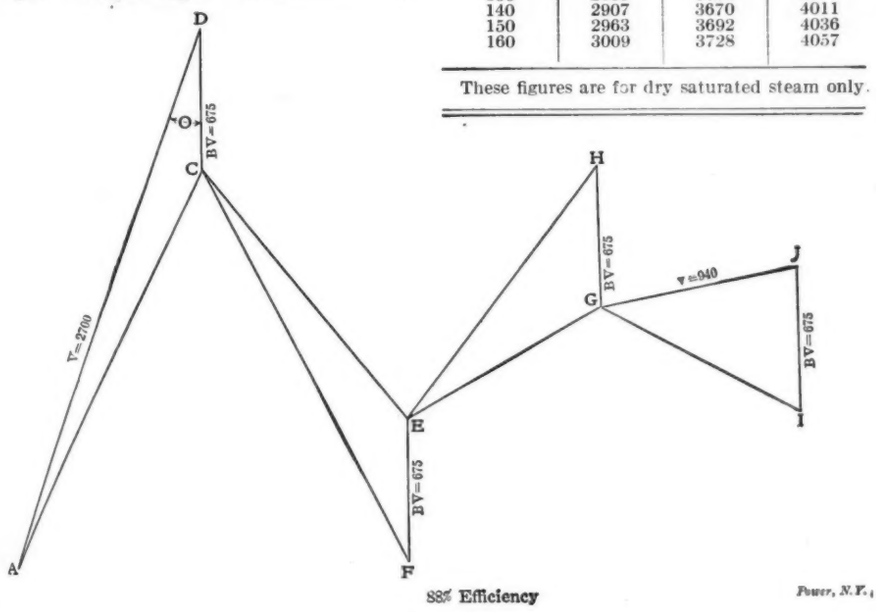


FIG. 5

Assume that this turbine is operating noncondensing upon a steam pressure of 100 pounds. Referring to the table, the nozzle velocity under these conditions will be found to be 2700 feet per second. The diagram, Fig. 3, shows in a graphical manner the action of the steam.

V = Velocity of steam at entrance to buckets.

BV = Bucket velocity (one-half steam velocity).

v = Velocity of steam at exit from buckets.

The line CD in length and direction represents the bucket velocity BV. The line AD in length and direction represents the absolute velocity V and the angle of the entering steam. This angle, O, as in Fig. 2, is made 20 degrees. Then the line AC in length and direction represents the velocity and angle of entrance of the steam relatively to the wheel. The line CF is made equal in length and angle, but of opposite direction, to AC; and FE, representing the bucket velocity BV, is made equal in length and direction to CD. Then the resultant CE in length and direction represents the velocity and angle of exit of the steam relatively to the wheel. Applying the efficiency formula,

$$E = \frac{V^2 - v^2}{V^2}$$

or

$$E = \frac{2700^2 - 940^2}{2700^2} = 0.88,$$

the efficiency is found to be 88 per cent. This is the highest efficiency practical with this type of bucket, as with more acute entrance angles it has been found impossible to get sufficient steam into the blades.

A turbine with such high bucket-wheel speeds would not, of course, be suitable for direct drive. In the De Laval turbine, which is an excellent example of this type, gearing has to be resorted to as a means of speed reduction to permit of their commercial application. In these turbines, where bucket velocities of 1270 feet per second are quite common, the rotative speeds range from 15,000 to as high as 25,000 revolutions per minute.

The rotative speed of a turbine can be decreased, within certain limits, by the use of bucket wheels of large diameter, with which a given peripheral speed can be obtained at a less number of shaft revolutions than in the case of a small-diameter wheel. When one considers, however, that a peripheral speed of 1350 feet per second requires a wheel 8 feet 7 inches in diameter for a rotative speed of 3000 revolutions per minute, the impossibility of doing much in the way of speed reduction along this line is at once apparent.

The diagram, Fig. 4, shows what happens when the bucket velocity is much less than one-half the steam velocity. In this case the velocity V of the entering steam is 2700 feet per second, as in Fig. 3. The bucket velocity BV, however, has been reduced to 675 feet per second, or one-quarter that of the entering steam. Applying the efficiency formula, as in the previous case,

$$E = \frac{2700^2 - 1500^2}{2700^2} = 0.68,$$

the efficiency is only 68 per cent. From this it will be seen that, in order to attain efficient steam utilization at practical rotative speeds, other methods than the above must be resorted to.

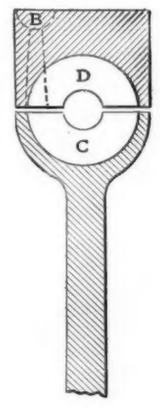
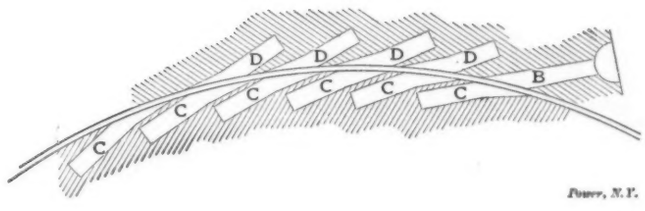


FIG. 6



Velocity Compounded Turbines—A very efficient method of obtaining moderate rotative speeds in the impulse turbine, and one that is commonly used in

the small sizes, is that of combining with a single pressure stage several velocity stages, or energy abstractions. This is accomplished either by returning the steam a number of times to the same bucket wheel, by means of a suitable reversing guide, or by passing it successively through two or more wheels, arranged side by side, with intermediate guides between. An arrangement of the

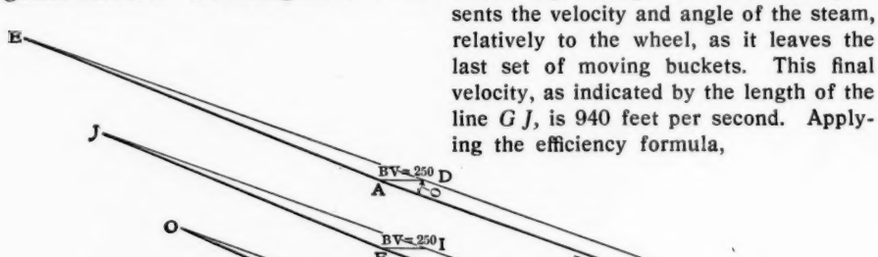


FIG. 7

latter kind is shown in either half of Fig. 9, in which *B* is the nozzle, *C* the bucket wheels and *D* the reversing guide or intermediate. The steam, entering through the nozzle *B*, where its pressure-energy is converted entirely into velocity, acts upon the first set of buckets, which abstract a portion of its energy and turn it into useful work. As the steam issues again from this first set of moving buckets it is caught by the stationary buckets of the reversing guide *D*, and redirected at a proper angle into the second set of moving buckets, where most of its remaining energy is abstracted and turned into useful work.

The action of the steam in this type of turbine is shown graphically in the diagram, Fig. 5. The line *A D* in length and direction represents the absolute velocity *V*, and the angle of the steam as it issues from the nozzle. The line *C D* in length and direction represents the bucket velocity *B V*. The angle *O* of the steam jet, relative to the plane of motion of the buckets, is 20 degrees, as in Fig. 2. Then *A C* in length and direction represents the velocity and angle of the entering steam, relatively to the wheel. The line *C F* is made equal in length and angle, but opposite in direction, to *A C*, and *F E*, representing the bucket velocity *B V*, equal in length and direction to *C D*. This bucket velocity, of course, remains constant throughout. Then the resultant *C E* in length and direction represents the velocity and angle of the steam, relatively to the wheel, as it enters the first set of stationary buckets. Also, *E H* is made equal in length and angle to *C E*, and *G H*, representing the bucket velocity *B V*, is made equal to *F E*. Then the resultant *E G* represents in length and direction the velocity and angle of the steam, relatively to the wheel, as it leaves the first set of stationary buckets and enters the second set of moving buckets. Next *G I* is made equal in length and direction to *E G*, and *I J* is made to represent the bucket velocity *B V*. The resultant *G J* in length and direction represents the velocity and angle of the steam, relatively to the wheel, as it leaves the last set of moving buckets. This final velocity, as indicated by the length of the line *G J*, is 940 feet per second. Applying the efficiency formula,

$$E = \frac{2700^2 - 940^2}{2700^2} = 0.88,$$

the efficiency is found to be 88 per cent., which is the same that was obtained in the previous case, according to the diagram of Fig. 3. From this it will be seen that by using two sets of moving, and one set of stationary buckets, the bucket speed can be reduced to one-quarter the steam velocity with as good efficiency as was obtained from a single set of moving buckets with a speed equal to one-half the steam velocity. The smaller sizes of the Curtis turbine are excellent examples of this type of machine.

Another method of velocity compounding that is frequently resorted to in small-size turbines is that of returning the steam, by means of a suitable reversing guide, two or more times to the same wheel. The bucket and nozzle arrangement of a turbine of this type is shown in Fig. 6. In this case the wheel is provided with buckets of the Stumpf type, a peculiar form derived from the pelton waterwheel. The steam enters through the nozzle *B* and is turned through an angle of 180 degrees by the semi-circular bucket. As the steam issues from the bucket it is caught by the stationary or reversing buckets *D*, turned through another half-circle, and redirected back into the rotating buckets *C*. This process is repeated as many times as there are reversing buckets to return the steam. Owing to the speed of the bucket wheel, the steam from the moving buckets *C* is each time returned to a different one of the reversing buckets *D*, passing through them progressively. When the steam has passed the last one of the reversing buckets it is allowed to escape into the exhaust. In this manner a large portion of the available energy of the steam is converted into useful work in an efficient way.

The diagram, Fig. 7, shows in a graphical way the operation of the steam in this type of turbine. Owing to the peculiar form of this bucket it cannot be diagramed with the velocity triangles

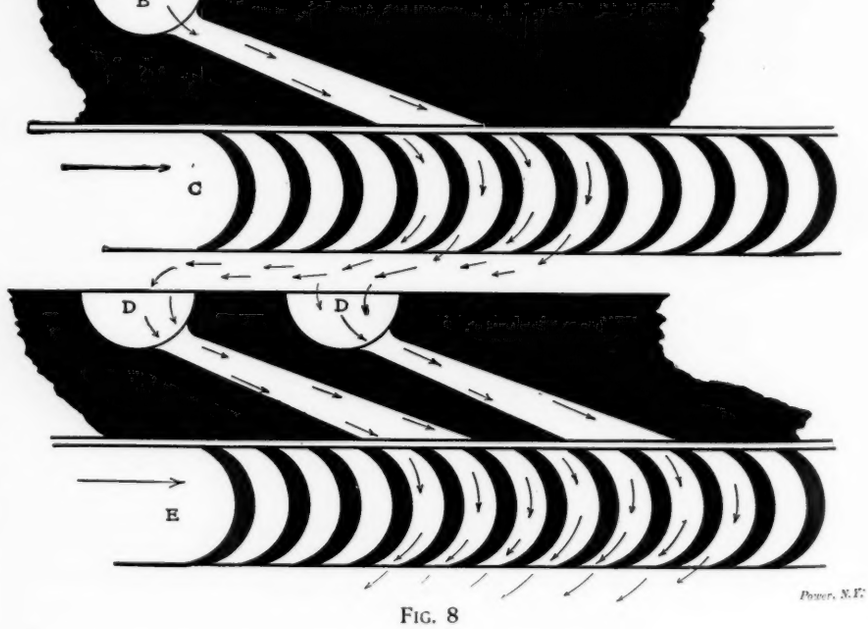


FIG. 8

that were used with the common form of vane, and it is therefore necessary to adopt the type of diagram shown. Referring to the diagram, the line *A B* in length and direction represents the absolute velocity *V* and the angle of the entering steam. As in the previous cases, *V*

is taken as 2700 feet per second and the angle O as 20 degrees. The bucket velocity $B\,V$, as represented by the line $A\,D$, has, however, been reduced to 250 feet per second. The line $B\,C$ is at right angles to $A\,B$, and $C\,D$ is parallel to $A\,B$. The line $D\,E$ is a continuation of $C\,D$, and equal to it in length. Then the resultant $E\,A$ represents in length the velocity V of the steam as it leaves the first set of rotating buckets. The line $F\,G$, made equal in length to $E\,A$, represents the velocity of the steam as it enters the first stationary or reversing bucket. Then the diagram is constructed, as in the previous case, but with the lines $F\,I$, $G\,H$, $H\,I$ and $I\,J$. Then the resultant $J\,F$ represents in length the velocity of the steam as it leaves the first stationary or reversing bucket. The line $K\,L$, made equal in length to $J\,F$, represents the velocity of the steam entering the moving buckets for the second time. Then the diagram is constructed, as in the previous case. The resultant $O\,K$ represents in length the velocity of the steam leaving the moving buckets for the second time. Then $P\,Q$, drawn equal in length to $O\,K$, represents the velocity of the steam entering the second stationary bucket. Again constructing the diagram, the resultant $T\,P$ represents in length the velocity of the steam leaving the second stationary bucket; $X\,Z$, made equal in length to $T\,P$, represents the velocity of the steam as it enters the moving buckets for the third time. Again, constructing the diagram, the resultant $W\,X$ represents in length the final velocity v of the steam as it issues from the moving buckets for the third and last time. This is found to be 360 feet per second. Applying the efficiency formula,

$$E = \frac{2700^2 - 360^2}{2700^2} = 0.98,$$

the efficiency is 98 per cent. This high efficiency of the Stumpf bucket is due to the steam being turned through an angle of 180 degrees, while in the other type it describes an arc of but approximately 150 degrees. In actual use, however, this type of bucket does not perform as efficiently as the diagram would seem to indicate, the steam being subjected to more friction, and also greater compression, than in the ordinary form of vane, so that its ultimate efficiency is little, if any, higher than that of the other type.

Examples of this type of impulse turbine are found in the Sturtevant and the Terry machines, and also in the German Elektra, although in this latter turbine a different form of bucket and reversing guide is used, but the principle is the same. These turbines, which are only built in the smaller sizes, are of moderate rotative speeds, and are therefore adapted to a wide field of commercial application.

Pressure Compounded Turbines—This type of turbine comprises two or more pressure stages, each pressure stage with a single bucket wheel, upon which the steam acts but once. It therefore requires no reversing guide. Fig. 8 shows the nozzle and bucket arrangement of a turbine of this type. The steam enters through the first-stage nozzle B into the first set of moving buckets C. The steam, however, is not expanded down to the back-pressure of the exhaust, as in the case of the single-stage turbine. Instead, it is expanded only enough to impart a portion of its total kinetic energy to the buckets. The nozzle expansions, or pressure drops, are so proportioned that the total kinetic energy is equally distributed among the total number of stages. By this method high velocity of the steam in the nozzles is avoided, which permits of its efficient utilization at very moderate bucket speeds. As the steam issues from the first-stage buckets C at reduced velocity, it passes into the second-stage nozzle D, where it is still further expanded, and its lost velocity regained. The steam then passes to the second-stage buckets E, where a further portion of its kinetic energy is abstracted. This process is repeated in each successive stage of the turbine, until practically all the available kinetic energy of the steam has been abstracted and turned into useful work. In this type of turbine, the greater the number of stages, the lower the rotative speeds that can be efficiently obtained. Cost of manufacture, however, places a practical limit on the number of stages that can be used in a commercial turbine. Generally speaking, this type of turbine has from 8 or 10 to 20 or 25 stages, according to size. The same diagram of the steam operation that was used with the single-stage turbine, Fig. 3, can be used in this case, each stage being considered as an independent single-stage turbine. Examples of this type of turbine are found in the Rateau, Zoelly, Holzwarth and Wilkinson machines. It is best adapted to the large and medium sizes, its cost of construction being practically prohibitive for small work.

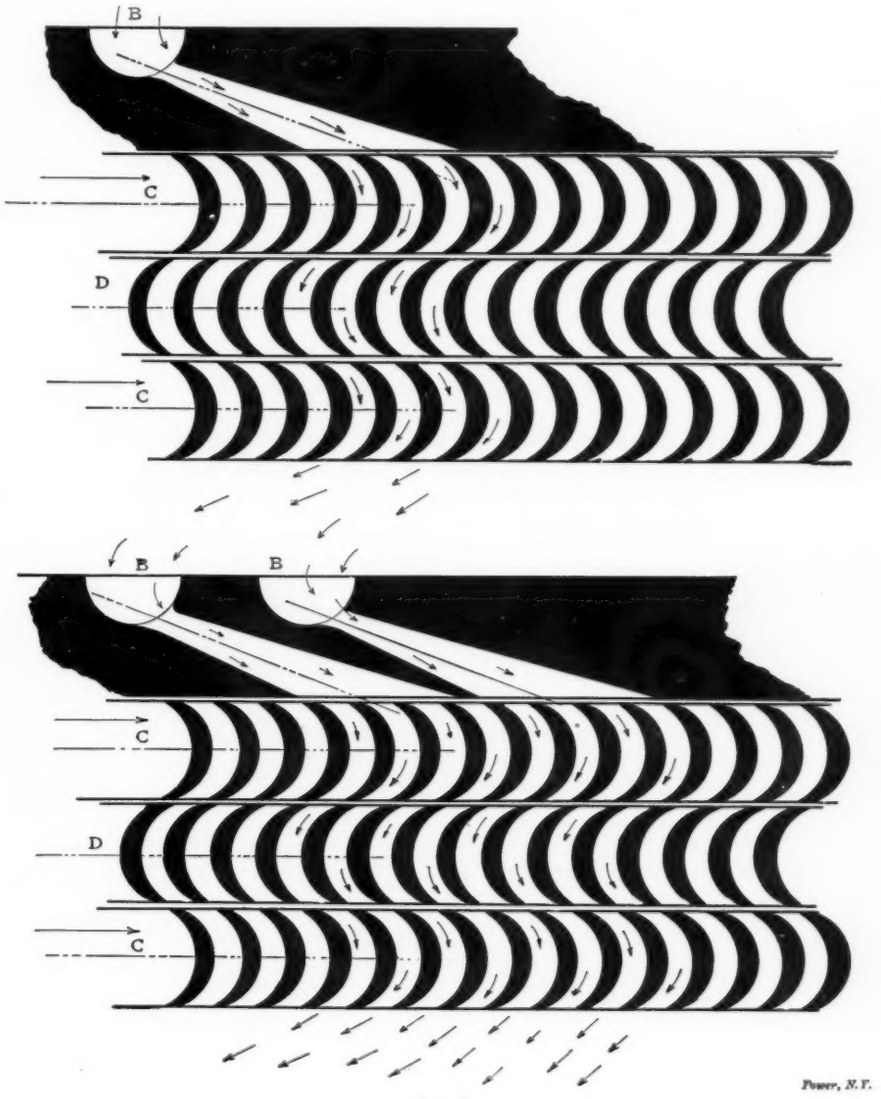


FIG. 9

Pressure and Velocity Compounded Turbines—Turbines of this type consist

of two or more pressure stages, each pressure stage comprising one set of nozzles and two or more rows of moving buckets, with intermediate guides. Fig. 9 shows the nozzle and bucket arrangement of a turbine of this type. The steam enters through the first-stage nozzle *B*, where it is expanded only sufficiently to convert a portion of its pressure into velocity. The steam then passes through the first set of moving buckets *C*, is caught by the stationary or reversing buckets *D* and redirected into the second set of moving buckets *C*. As the steam issues from this last set of first-stage buckets, it passes into the nozzles of the second stage. Here it goes through a similar process, and so on, throughout the successive stages. The same diagram of steam operation that was used with the single-stage turbine, Fig. 5, can be used in this case, each stage being considered as an independent single-stage turbine.

For a given rotative speed this type of turbine requires a less number of stages than the previous type shown in Fig. 8, as the pressure drops per stage are much greater. In commercial form it usually has from two to six stages, according to size. An excellent example of this type of impulse turbine is found in the larger-size Curtis machines, in both the horizontal and the vertical form.

Comparison of Steam and Hydraulic Turbines

A turbine may be defined as a wheel, provided at its periphery with paddles, buckets or blades, against which a moving fluid, such as water, gas or steam is directed in such manner as to cause a rotary motion. The waterwheel is a simple turbine of the Branca type. In considering the steam turbine this similarity to the waterwheel should always be borne in mind, and it should be carefully disassociated from the reciprocating engine, with which it has little similarity. The common mistake of trying to understand the steam turbine from the reciprocating-engine point of view is accountable for many erroneous ideas regarding this new type of prime mover.

The general principle of operation of the motive fluid is the same in both the hydraulic and the steam turbine. In fact, most of the present theories and formulas for the steam turbine were originally derived from standard hydraulic-turbine practice. Hydraulic turbines utilize the available energy in a head of water in different ways, according to their type. The overshot wheel depends upon the weight of the water for its power; the undershot wheel and the pelton wheel use its velocity, or impulse; some hydraulic turbines operate by the pressure of the water; others by a combination of all these methods. For comparison with the steam turbine the pelton wheel is selected, as its principle of operation is identical with that of the single-stage steam turbine of the impulse class. The design of the single-stage impulse steam turbine differs from that of the pelton wheel only as regards velocities and volumes of the working fluid. This will be readily seen from the following comparative calculations for both steam and water.

The power of a fall of water is the product of the weight of water discharged in a given unit of time into the total head. If Q = Cubic feet of water discharged per second, D = Weight of a cubic foot of water (62.36 pounds at 60 degrees Fahrenheit), H = Total head in feet, then

$$D\,Q\,H = \textit{Total power in foot-pounds per second,}$$

and $D\,Q\,H$ divided by 550 (the mechanical equivalent for horsepower in foot-pounds per second) gives the total horsepower in the water. For example, assuming a head of 50 feet, and a discharge of 25 cubic feet per second, the total horsepower would be,

$$\frac{62.36 \times 25 \times 50}{550} = 141.7 \textit{ horsepower.}$$

If V = the velocity of efflux, h = the head in feet, then

$$V = \sqrt{2gh} \text{ or } V = \sqrt{2 \times 32.16 \times 50} = 56.7$$

feet per second, which is the mean velocity of efflux. The kinetic energy available in the water would be,

$$141.7 \times 550 = 77{,}935 \textit{ foot-pounds.}$$

In a steam turbine, with steam at 160 pounds initial pressure, expanding through a nozzle down to atmospheric pressure, the velocity of exit from the nozzle would be 2970 feet per second. If V = the velocity in feet per second of steam discharged from the nozzle, then the kinetic energy of the jet would be,

$$K.E. = \left(\frac{V}{8}\right)^2 \text{ or } \left(\frac{2970}{8}\right)^2 = 137{,}641$$

foot-pounds per second per pound of steam. The total horsepower in one pound of steam under these conditions is,

$$\frac{137{,}641}{550} = 250 \textit{ horsepower.}$$

From this it will be seen,

$$\frac{141.7}{250} = 0.565,$$

that approximately half a pound of steam per pound, at an initial pressure of 160 pounds, equals in available energy 25 cubic feet of water per second with a 50-foot head. The volume of one-half pound of steam expanded down to atmospheric pressure is 13.18 cubic feet, or approximately one-half that of the water. The steam, however, has a velocity of 2970 feet per second, while the velocity of the water is only 56.7 feet per second. Therefore, the cross-sectional area of a suitable passage for handling this amount of water at this velocity would be,

$$\frac{13.18 \times 2970}{25 \times 56.7} = 27.5,$$

or 27.5 times greater than that required for handling the steam.

Commercial Application of the Turbine

The steam turbine finds its greatest field of usefulness in the driving of electric generators. In this class of service high speed is very desirable from the electrical point of view, increasing the efficiency of the apparatus, decreasing its weight, and the space occupied, and producing a more uniform turning moment than is possible with the reciprocating engine. For these reasons the steam turbine is rapidly coming to the front for central-station work in railway and lighting service.

The steam turbine is also proving itself highly desirable for centrifugal pump work, especially for the modern high-pressure pumps for fire service and boiler-feed work, where pressures of 150 pounds and upward are quite common. The speed of such pumps is above the field of the engine and the electric motor, if direct connection is to be attained, and the turbine finds a ready field in this class of work.

For the driving of blowers, and especially since the advent of the so-called "kinetic blower," which delivers air at pressures of from five pounds upward, with proportionately high rotative speeds, the turbine is unexcelled. In gas-plant and foundry work it has a decided advantage over the electric motor, in that it is not affected by heat and dust, and can therefore be placed conveniently near the furnaces without injury.

For marine service the turbine claims important advantages in the matter of large power in a small space, light weight, freedom from vibration and increased speed of the boat over that possible with the reciprocating engine. The steam turbine, like the reciprocating engine and the gas engine, has its own particular field to which it is best adapted, and which no other prime mover will fill equally well.

John G. Neibert, of Hagerstown, Md., who is chief engineer of a local grain elevator, recently had a very narrow escape at the hands of three robbers. Soon after he had started the fire under the boiler, three men, said to be negroes, entered the room and attacked him. He is a powerful man, and for a time kept them off, but was finally overpowered and robbed of $10, all the money he had with him. His assailants then tried to hide their crime by cremating him. Only his head could be forced through the furnace door, and this attempt was abandoned. The robbers then placed Neibert between the spokes of the flywheel, and set the engine in motion. At the first revolution the engineer slipped to the floor, where he was found still unconscious some time later.

Unusual Method of Erecting 90-Foot Stack

BY WARREN O. ROGERS

A few days ago, while passing a manufacturing establishment, I saw, laying on the ground, three sections of a new iron stack. Being of an inquisitive turn of mind I ambled into the office and ascertained that the men engaged to erect the stack were already on the job. Thinking it would be interesting to watch and describe the different stages in the process of erection, the camera was loaded and photographs taken as the work progressed.

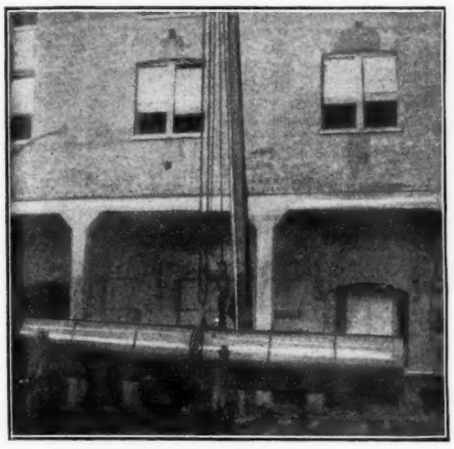


FIG. 1. SECTION OF STACK JUST CLEARING THE GROUND

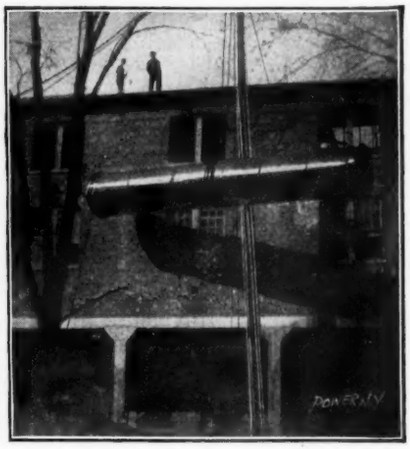


FIG. 2. SECTION OF STACK NEARING ROOF OF BUILDING

The work of erecting the stack was somewhat out of the ordinary, as it was necessary to first hoist the three sections to the roof, take down the old and smaller stack, and hoist the new one onto the same stack foundation, which extended up to the second floor of the building, but on the inside of the end wall of the building proper. This necessitated lowering the new stack down through the roof, as well as hoisting the old stack up two stories so as to clear the roof with its bottom end.

The first work necessary was that of placing the 65-foot, hexagon pole in a vertical position beside the building, as shown in Fig. 1. This pole, which was 8 inches through at the top and 14 at the bottom, was fitted with four guy ropes at the top; a 1-inch rope was also wound six turns around the top end, on which a block and tackle was hooked. The rope rove through the blocks was 1 inch in diameter. Another block and tackle was also hooked onto the same winding of rope for hoisting a man in the boatswain chair, in case the tackle became choked or other matters required attention at the top of the pole.

The rope holding the top blocks was prevented from slipping by cleats nailed to the pole under the rope winding. When these preparations had been made the guy ropes next to the building were passed up over the roof and by means of a windlass set in a frame, the pole was "up-ended," the two side guides being kept taut to prevent the pole from tipping sidewise. The base of the pole rested on planking to prevent it from sinking into the ground when the load of the stack was applied. When the pole reached a vertical position, the guy ropes were secured to trees and other handy objects, such as beams in adjoining buildings, the guy rope passing in through open windows.

A capstan was next secured to the ground at a distance of about 30 feet from the base of the pole, and the loose end of the hoisting tackle wound around the capstan head. As the capstan was turned the rope from the block at the top of the pole wound up on the capstan. From the top block to the capstan the rope passed through a single block secured to the bottom of the pole, so that the pull on the capstan was at a right angle to the pole.

These details being completed, the top part of the stack, composed of six sections each 5 feet long, riveted together, was rolled into position at the foot of the pole, and seven turns of 1-inch rope wound tightly around the center of the stack section, a U-link being used so that in the winding process the link was bound tightly to the stack without the trouble of passing the end of the rope through a ring at each wind, which would have been necessary had a ring been used.

The two ends of the U-link were fitted with a bolt and nut, making it impossible for the ring on the bottom block to become detached. This method was also much safer than if a common hook were used under the rope coil, as it was impossible for it to work loose.

Before hoisting, guy ropes were secured to each end of the stack section to hold it steady, the loose ends of these ropes being held by men, whose business it was to prevent the wind swinging the stack against the building. Fig. 1 shows the first section of stack just as it had cleared the ground. The guy ropes at the end are shown, while on the stack is seen one of the riggers, whose duty it was to watch the block to prevent jamming of the rope, and also to watch for any other trouble that might occur.

In this instance the stack was raised by means of a horse hitched to the capstan pole, walking round and round in the same manner as one sometimes sees house moving in country places.

FIG. 3. SWINGING STACK ONTO ROOF

FIG. 4. SAFELY LANDED

Fig. 2 shows a section nearing the roof of the building, the balance being perfect, which makes the work of hoisting

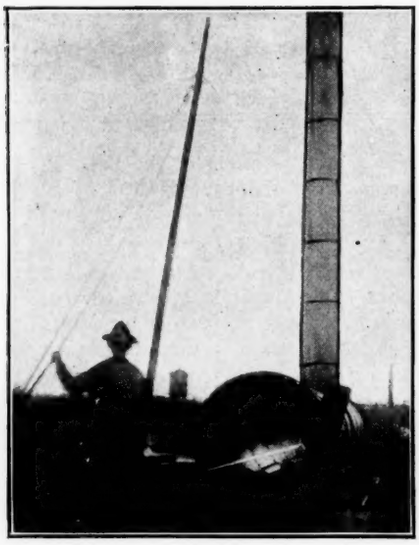


FIG. 5. FIRST GIN POLE NEARLY UP

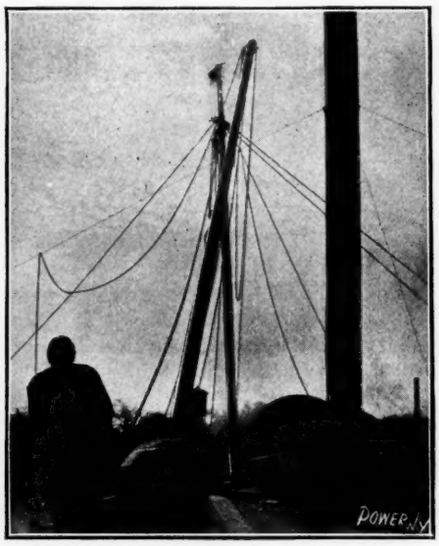


FIG. 6. PLACING THE HEAVY GIN POLE

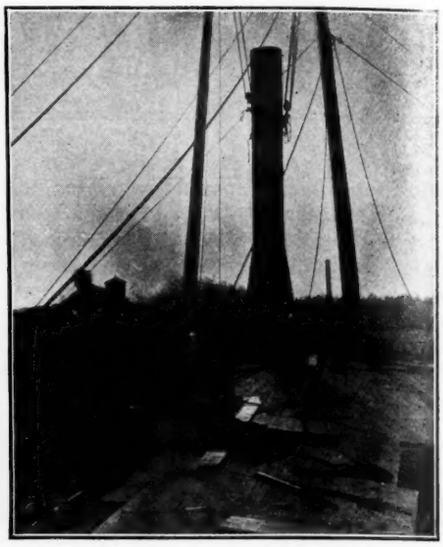


FIG. 7. REMOVING BOTTOM SECTION OF OLD STACK

the section into the place less arduous than if it were hung on an angle. Fig. 3 shows the nearness of the tackle blocks as a section of the stack is being swung into place, showing that some space should be figured on to allow for twisting the ropes, as shown in Fig. 4, in order to swing the stack from the outside to the inside of the pole onto the roof.

After the section was hoisted to the proper hight, one end was swung in on the roof, as shown in Fig. 3, and lowered when the position shown in Fig. 4 was reached; the other two sections were safely hoisted to the roof in the same manner.

The next procedure was to hoist the first gin pole on the roof. This consisted of a small pole, 6 inches in diameter at the base and about 25 feet high. It was fitted with four guy ropes, a heavy hoisting tackle and a lighter one for a "boatswain chair." Getting this pole into an upright position was the most difficult part of the work in erecting the stack, because there was nothing with which to lift it. This pole was placed crosswise with the roof and guy ropes were carried out in four different directions, a man being stationed at the loose end of each to steady the pole as it assumed an upright position. To prevent the base of the pole from slipping, one end of a block and tackle was attached to it, the other being secured to a beam.

The hoisting tackle was then attached to a roof beam by securing it to several links of rope passed over the edge of the roof of the building and in through a window. Then, with a man tightening up on the base block, one man stationed at each guy rope, several more on the loose end of the lifting block and two men to lift the smaller ends of the pole, all was ready to hoist the pole to a vertical position.

When the smaller end of the pole was raised the tackle exerted a pulling force similar to that exerted on the crank pin of an engine, with the crank off the center. In order to ease the strain on the hoisting tackle, the small end was raised to an angle of about 25 degrees, when the purchase of the lifting tackle was enough to easily hoist the pole into position. When the guy ropes were made secure at their outer ends. Fig. 5 shows the first gin pole nearly in an upright position.

The heavier working gin poles were next hoisted into position, Fig. 6 showing the manner of doing this particular piece of work. Two were used, each rigged out with four guy ropes, hoisting falls and blocks and a boatswain chair rig. Cleats were nailed to the top of the pole under the rope coils to which the falls were attached to prevent the rope from slipping. When all was ready the pole was hoisted into position and the guy ropes secured. The first gin pole was then worked over to the other side of the stack where the second heavy pole was to be set. This was done by loosing up on the guy ropes, but keeping them tight enough to balance the pole. When in position the second pole was hoisted into place and the guy ropes secured. No attempt was made to balance the poles in hoisting, because the nearer vertical they hung when suspended, the better, as that was the position they were to eventually assume.

In Fig. 7 are shown the two gin poles in position, the bottom section of the old stack being removed from its foundation, and just clearing the roof. The method of securing the block and tackle is shown, there being two separate windings of rope for each block attachment. The U-link is used, and the block hook engages the bolt and screw.

As the bottom section of the new stack was made of the same diameter as the old, it was a simple matter to attach a falls to the bottom section and lower it into place, when it was plumbed and the anchor bolts tightened up. The top of the old stack was removed in one section by

FIG. 8. HOISTING TWO TOP SECTIONS OF STACK

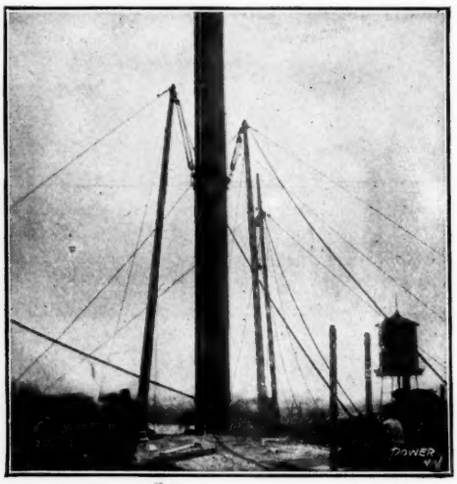


FIG. 9. ALMOST READY FOR USE

the same method, as shown in Fig. 7, the block being secured to the stack at a point somewhat above the center.

Before lifting the upper portion of the new stack into place, the two sections were bolted together, flanges having been provided for that purpose. The stack was then attached to the blocks in the manner already described, and, as an extra precaution, a saddle was made, consisting of lengths of rope passing from one block hook, under the stack, to the other to prevent the block and lashings from shifting on the stack.

In raising the new sections of stack, each block and tackle was worked by means of a separate windlass, the rope passing through a single block secured to the base of each pole. Fig. 8 shows the top two sections being hoisted into place.

In Fig. 9 is shown the stack with the upper section just clearing the bottom one, ready to be lowered to place. When this was done the sections were bolted together, the guy ropes, which had been secured to the band around the stack before hoisting, were secured at their outer end, the tackle slacked off and the new stack was ready for business.

There is more to stack raising than the attention to rigging and lifting parts, because a certain amount of skill is required, and good judgment must be used in determining just when and how to attach the rigging, besides deciding as to whether certain falls are sufficiently strong to withstand the load.

It is doubtful if the ordinary engineer, without experience in stack raising, would be able to accomplish the work in a successful manner within a reasonable time, especially with a large stack. With ordinary small stacks the method described in the foregoing would doubtless be of assistance, and modification of the practice could be applied to advantage. One thing to be particular about is to know that the block and tackle is sufficiently strong to withstand the weight which will be suspended from it.

Operation of Boiler Feed Systems

Pertinent Points to Be Considered in the Design and Operation of Feed-water Systems for Modern Condensing and Noncondensing Plants

BY WILLIAM F. FISCHER

The design of a system of boiler-feed piping for the modern high-pressure power plant is a matter of considerable importance, and should be carefully worked out in detail before installing the usual auxiliary apparatus. In most cases the pumps and heater can be located to much better advantage after the

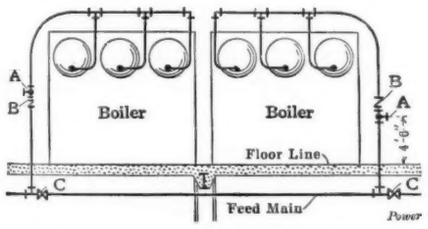


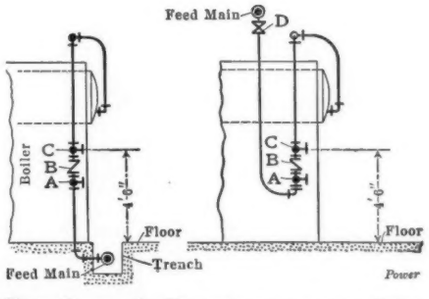
FIG. 1. LOCATION OF FEED MAIN

general arrangement of the feed-piping system has been decided upon.

In plants of medium size, the arrangement of engines and boilers is much the same throughout, the designer usually following one or two general schemes. In one case the engines and boilers are placed back to back, with a division wall placed between, dividing the boiler and engine rooms. This arrangement is more often employed than any other as the steam piping between the boilers and engines is considerably shorter than where the boiler and engine rooms are placed end to end. In modern steam plants of over 1000 horsepower, engineers frequently specify duplicate feed mains as, interruptions to the boiler service may frequently be traced to trouble in the feed-water piping. The most important point to be considered is that of reliability of service; therefore the system should be so arranged that at no time will it be necessary to shut down more than one battery of boilers, should a break occur at any point in the feed main or branch pipes.

RING SYSTEM

In this respect the "ring system" of piping may be employed to good advantage, having valves placed in the main between each inlet and outlet branch, to or from the main. Should a break or blowout occur in any section of the main, or in any branch pipe, the valves in this section can be closed, and the feed water carried around the ring in the opposite direction, thus keeping the



FIGS. 2 AND 3. UNDERGROUND AND OVERHEAD FEED LINES

rest of the boilers in operation without interruption.

The cost of additional piping, valves and fittings required for either the "double main" or the "ring system" is ordinarily but a small portion of the total cost of the plant, and if continuity of service is desired, this extra cost should not be considered, except in very small plants. The main argument against a duplicate main system is perhaps the fact that one-half of the piping is lying idle most of the time.

At times boiler-feed pipes may be subjected to heavy pulsations caused by the feed pumps.

RESERVE FEED PUMP

No plant of any consequence should depend on one pump for its supply of feed water. At the very least, two boiler-feed pumps should be provided for, then, in case one fails, the reserve pump can be put into service while making repairs.

If the extra cost prohibits the installation of a reserve pump, an injector should be installed to take its place. In many cases both the injector and reserve pump are installed to guard against any possible interruption to the boiler service.

LOCATING THE FEED MAIN

Where a basement is provided beneath the boiler house for the removal

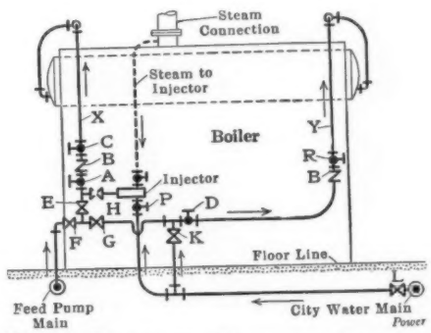


FIG. 4. PIPING FOR PUMP AND INJECTOR

of ashes and the accommodation of forced-draft apparatus, air ducts, etc., a space may be partitioned off and used as a pump and heater room, where the feed pumps, heater and other apparatus may be located. The boiler-feed main

may also be placed in the basement and suspended from the floor beams above, having branches rising through the floor to each battery of boilers as indicated in Fig. 1. The feed-water regulating valve *A* should always be a globe valve, preferably a brass or bronze valve with a regrinding disk. A gate valve cannot be closely regulated and when open part way often chatters, owing to the pulsations of the feed pumps. The check valve *B* should have either a brass or bronze body valve of the swing-check type with regrinding disk. The regulating valve *A* should, in all cases, be placed within easy reach of the boiler attendant (about 4 feet 6 inches above the boiler-room floor line), and preferably on the same side of the boiler as the water column.

The feed main is divided into sections by placing a gate valve *C* in the main between each branch to or from the main as mentioned above.

Where there is no basement below the boiler-room floor the feed main may be run in a covered trench alongside or in front of the boilers and a branch pipe carried up to each battery of boilers, as shown in Fig. 2. This trench should be of sufficient width to permit of repairs being made on the feed main, when necessary. The trench should pitch in one direction and be provided with a drain to keep it free of water.

Sheet-iron or cast-iron cover plates should be provided the entire length of the trench, each of such length as to be easily raised by one man.

If the trench shown in Fig. 2 is undesirable, the feed main may be run overhead, as shown in Fig. 3, and a branch pipe carried down the side of each boiler, bringing the feed valves in an accessible position from the boiler-room floor, as shown.

If for any reason it is necessary to place the feed-regulating valves either above the boiler or in other inaccessible positions they should be provided with either a return-extension stem, a system of gearing, or a chain and chain wheel, so arranged as to be operated from the boiler-room floor. In no case should the attendant be compelled to climb a ladder in order to operate the regulating valve. This is a point sometimes overlooked by the designer, and causes the boiler attendants needless trouble every time the valve requires adjustment.

Arrangement of Valves

The simplest arrangement of feed and check valves in boiler branch pipes, is that shown in Fig. 1. Here but two valves in each branch are placed between the boilers and the feed main. At the right of Fig. 1 the check valve *B* is shown placed above the feed-regulating valve *A*. In this case the regulating valve *A* is used as a stop valve when cutting out the boiler.

This arrangement, although quite common in small plants operating intermittently, is not to be recommended, as any sticking or leakage of the check valve necessitates shutting down the boiler while repairing or regrinding the valve disk, and to remove the regulating valve *A* means (with a single feed main), a temporary shutdown of a large portion of the main.

To the left of the boilers the stop and regulating valve *A* is shown above the check valve *B*. This method should never be employed as the check valve cannot be removed without shutting down the boiler and part of the feed main.

A much better, and more common method of arranging the valves, is shown in Fig. 2, where the check valve *B* is placed between the stop valve *C* and the regulating valve *A*. In this case the check valve can be removed for repairs, keeping pressure on the boiler and feed main by closing valves *A* and *C*.

If screwed-end valves are used, a flanged or screwed union should be placed in the line between the check valve and one of the valves *A* or *C* so as to be able to remove the check valve.

The regulating valve *A* may be set for the required flow of feed water and left undisturbed by using the stop valve *C* when cutting out the boiler.

With the valves arranged as in Fig. 3, both the check valve *B* and regulating valve *A* may be removed from the line, keeping pressure on the boiler and feed main by closing valves *C* and *D*. Valve *D* is placed next the feed main and should be a gate valve so as to reduce the friction in the line.

This arrangement of valves is particularly desirable in stations where the boilers are operated continuously, because the regulating valve *A* is subjected to severe usage. As the stop valves *C* and *D* are either run wide open, or closed tightly, they are subject to much less wear than the regulating valve *A*.

The check valve *B* is absolutely necessary in each boiler branch for several reasons. It prevents the water in the boiler from backing up in the feed main, should the pumps slow down, and should the feed main break at any point it will prevent the boiler from emptying itself through the ruptured section.

Pump and Injector Feeding System

In Fig. 4 is shown a good arrangement of piping and valves at the side of each battery of boilers, where provision is made to feed with either pumps or injector, or both. When feeding from the pumps the valves *G, H, D* and *P* should be closed and valves *A, C, E* and *F* opened, using valve *A* as the regulating valve.

When feeding from the city water main, through the injector, valves *E, G* and *K* should be closed and valves *A, C, H, P* and *L* opened.

When feeding with the pumps and injector at the same time, valves *E* and *K* should be closed, and all other valves opened wide, thus permitting the feed water from the injector to enter the boiler through pipe *X*, at the left of the diagram, and the feed water from the pumps to enter at the right through pipe *Y*, using *A* and *D* as the regulating valves.

In piping the injector, steam should be taken from the highest point and direct from the boiler (not out of the steam main), because the injector demands dry steam at full boiler pressure.

When washing out or flooding the boilers with city water under pressure, valves *P, F* and *H* should be closed and all other valves opened wide, thus flooding the boiler through both branches *X* and *Y*.

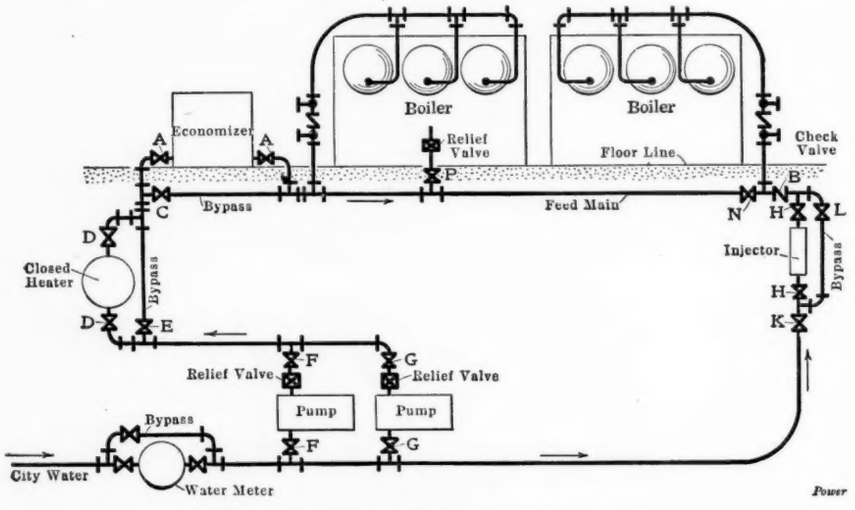


Fig. 5. Piping for Noncondensing Plant

Piping Systems for Noncondensing Plants

A system of boiler feed-water piping for a noncondensing plant, closed type of heater and economizer, is shown in Fig. 5.

The boiler feed pumps draw their supply from the city mains or other convenient source, and force it through the closed heater and economizer into the feed main and from thence to each battery of boilers. A reserve pump is provided, which may be connected up and cut out of service until needed, by closing valves *G G*.

A meter for registering the supply of city water is shown located on the suction side of the pumps and provided with a bypass to cut the meter out of service for repairs or adjustment. In large stations two or more meters are installed, and either may be cut out of service.

The operation of the system is as follows: By opening *A, D, F* and *N* the feed water is pumped through the closed heater and economizer and feed main, when it enters are boilers through each branch pipe.

To cut out this heater for repairs close valves *D D*, and open *E* in the bypass. To cut out the economizer close valves *A A*, and open valve *C* in the bypass.

The stop valve *N* in the feed main divides it into two sections for convenience in making repairs.

By closing valves *C, A* and *L* and opening valves *N, H* and *K* the boilers may be fed from the injector. To bypass the injector and flood or wash out the boilers with city water (if under pressure), close valves *H H* and open *L*.

The diagram, Fig. 6, shows a feed-water system for a noncondensing plant using an open feed-water heater. In this case the heater is located on the suction side of the pumps, which, receiving their supply of hot water from the heater, force it through the water meter and economizer to the feed main and from there into the boilers. The feed-water heater, economizer and water meter are each provided with a bypass.

The open feed-water heater should be set at least 2 feet above the pump suction valves, so that the hot water will gravitate to the pump and raise the suction valves from their seats. This is necessary, as the feed pumps will lift hot water successfully unless specially constructed, as hot water gives off a vapor when the pump plunger creates a vacuum in the suction pipe.

The cold-water supply should gravitate to the heater, or enter it under pressure, and the supply should be controlled automatically by a float valve within the heater.

A separate injector main is provided as shown in the diagram making connection to each boiler branch pipe. With this arrangement any boiler or boilers may be fed through the injector main while the others are being fed through the pump main, a feature impossible with the single feed main shown in Fig. 5.

In Fig. 6 the water meter is shown on the pressure side of the pumps and provided with a bypass.

All high-pressure drips and returns from the heating system, where one is used, can be piped to the open heater and returned again to the boilers with the feed water.

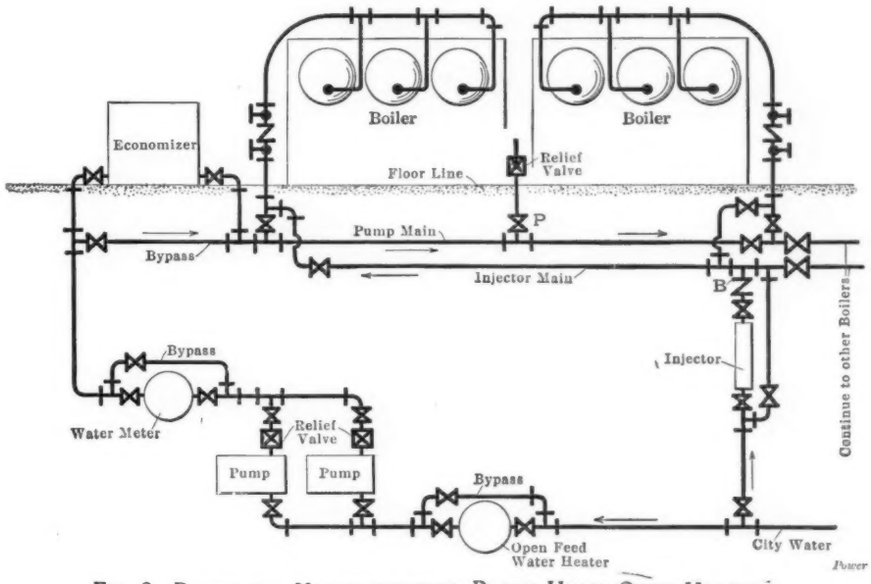


FIG. 6. PIPING FOR NONCONDENSING PLANT USING OPEN HEATER

AUTOMATIC RELIEF VALVE

An automatic feed-water relief valve is shown above the boiler-room floor in Figs. 5 and 6. The stop valve *P* is placed next the feed main for use when repairing the relief valve.

The function of the relief valve is to prevent excessive pressure in the closed heaters, economizers and feed main in case all the feed valves are accidently closed before shutting down the feed pumps. For the same reason a safety valve or a safety-valve connection is usually provided on all closed feed-water heaters and economizers.

By placing the relief valve above the boiler-room floor line the attendant's attention is attracted by it when it starts to discharge, thereby notifying him of the excess pressure in the feed main. If any of the bypass valves are accidently closed, however, shutting off the supply of feed water before it reaches the main, the relief valve is useless.

The modern practice in the design of medium- and large-size plants is to place a relief valve in the discharge pipe immediately adjacent to each feed pump, as indicated in Figs. 5 and 6. In this position, if in proper working order, the relief valves will prevent accidents from excess pressure, no matter in what part of the feed main the valves are closed.

PIPING SYSTEM FOR CONDENSING PLANT

A system of feed-water piping for a condensing plant is shown in diagram in Fig. 7. The plant is supposed to contain two engines, each with an independent vacuum heater, surface condenser, hotwell pump and air pump. There are also two boiler-feed pumps, two auxiliary exhaust-steam feed-water heaters of the closed type, and an economizer.

The closed heaters receive their supply of exhaust steam from the feed pump, condenser pumps and other auxiliaries not shown in the diagram. The exhaust from the main engines passes through the vacuum heaters before reaching the condensers. The steam condensed in each condenser is pumped to a hotwell, from which the feed pumps draw their supply.

If the hotwell is placed above the pump chamber, the hot water gravitates to the pumps, as already mentioned under noncondensing systems. If the feed pumps lift their supply from the hotwell, they should be specially constructed to lift hot water, otherwise much trouble may be experienced in getting the proper amount of feed water to the boilers.

A fresh-water connection should be made from the city water main to the hotwell for replenishing the water from the condensers and supplying fresh water when needed. This supply should be controlled by a float valve which opens automatically when the feed pumps draw water from the hotwell faster than it is discharged from the condenser pumps. The stop valve *U* is provided for cutting off this supply when repairing the float valve.

A connection is also made between the city water main and the pump suction through the valve *T*, and the city water line continued to the injector. The water meter placed at the pressure side of the pump should be capable of measuring hot water. When the feed pumps

are drawing their supply direct from the city water main, the valves *S* and *U* are closed and valve *T* opened. Either of the pumps may be cut out of service by closing valves *P P* or *R R*, and held in reserve until required.

The water meter, feed-water heaters, economizer and injector should be bypassed, as shown.

Operation of System

By closing valves *O, M, F, L, C, W* and *X*, and opening valves *N, K, H, G, E, D, A* and *Y*, the water from the feed pumps is forced through the water meter, vacuum heaters, auxiliary steam feed-water heaters and the economizer into the feed main and from there to the boilers. To bypass the water meter, open valve *O* and close valves *N N*. To bypass the vacuum heaters, close valves *K, G* and *H*, and open *L*. To bypass the auxiliary heaters, close valves *D* and *E*, and open *F*, and to bypass the economizers, close valves *A A*, and open valve *C*.

When pumping direct from the feed pumps to the feed main, use the tie line by closing valves *K, L, C, A, W* and *X* and open valves *O, M, Y;* this gives a more direct run for the feed water. When feeding with the injector, close valves *U, T, A, C, M* and *X* and open valves *V, W, Y*. A check valve *B* should be located in the discharge pipe from the injector, and a stop valve *Y*, located in the feed main, between each boiler branch outlet, thus dividing the feed main into independent sections.

If the hotwell is not provided with some suitable material for filtering the oil from the condenser-pump discharge, a filter should be placed between the feed pumps and the boilers.

It would be a much better arrangement, perhaps, to have an independent main from the injector to the boilers, as shown in Fig. 6, so that water from either the injector or boiler feed pumps could be supplied to any boiler, independent of how the others are fed. The pumps in this case could be equipped with relief valves on the discharge end, as shown in Figs. 5 and 6.

If a live-steam purifier is used, it should be placed above the boilers. The feed water, after it has passed through the other heaters or the economizer, enters the purifier and after being subjected to live-steam treatment gravitates to the boiler.

The tie line shown above the boilers, Fig. 7, connects the two feed branch pipes, and might be useful in feeding across from one battery of boilers to the other while grinding or repairing the check or regulator-valve disk. A stop valve should be provided in this tie line, as shown.

Material for Boiler-feed Mains

Feed-water piping may be put together with either screwed or flanged connections, depending on the size of the mains and the conditions to be met. The fittings are either of brass or cast iron. To reduce friction in the mains, all bends and elbows should be of long radius, and gate valves should be used in preference to globe valves, except at the boilers where the regulating valves should in all cases be globe valves with regrinding disks, preferably brass-body valves.

Experience has shown that hot feed water corrodes wrought iron and steel pipe very rapidly, due to certain impurities in the water. It is therefore desirable to use what is known as "extra-strong" or double extra-strong pipe for this purpose. Even the double extra-strong pipe may require replacing in a few years' time, however, and it is therefore more desirable to use brass pipe, where the cost is not too great, as brass pipe is not subject to corrosion.

Galvanized-iron pipe has been successfully used in many cases. Some engineers prefer making the feed lines of cast iron and brass throughout; that is, piping 4 inches and larger in size is made of cast iron, and the smaller sizes, such as boiler and pump branches, of brass. Cast-iron pipe is treacherous, however; and should be used with caution.

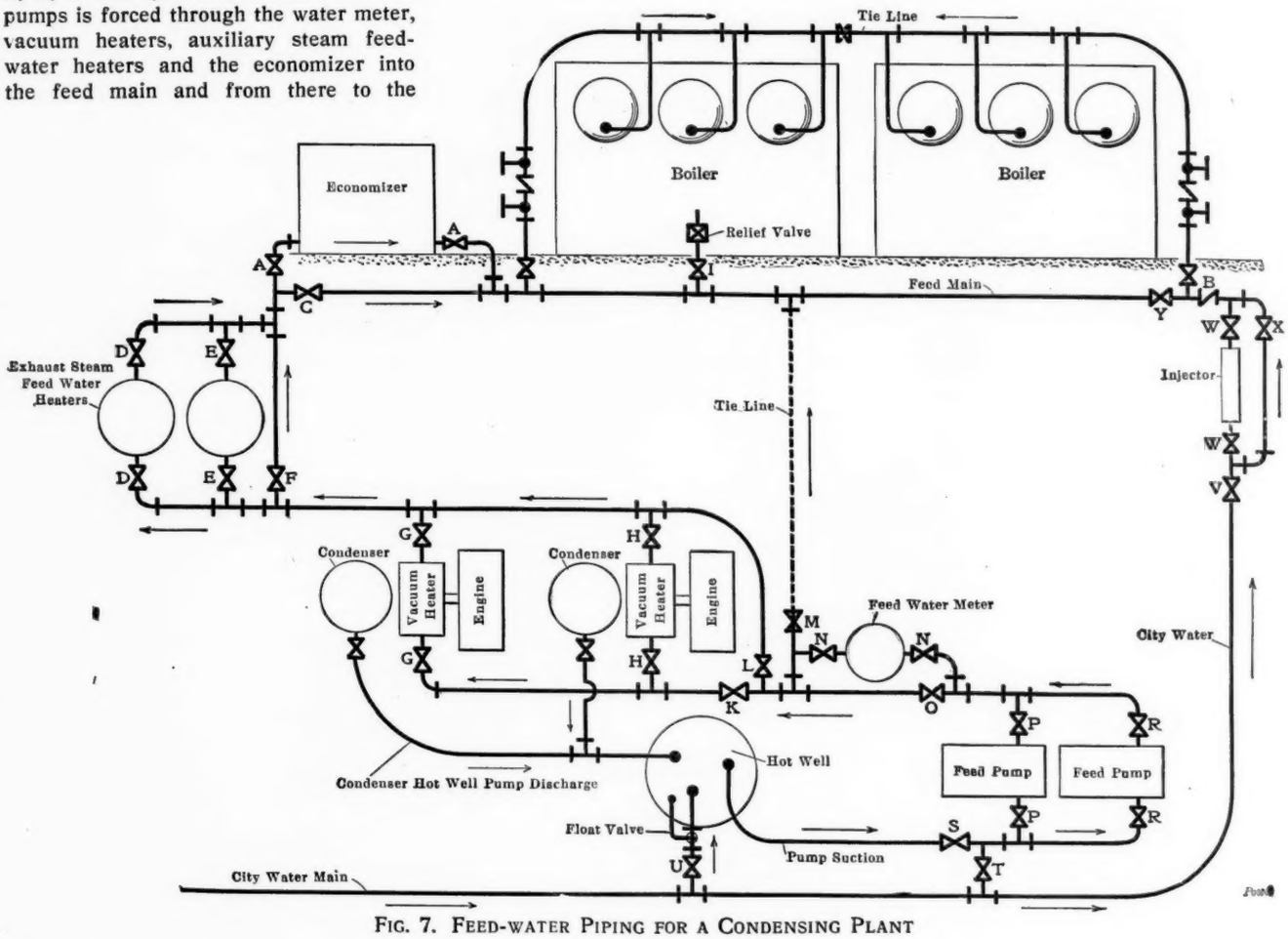


Fig. 7. Feed-water Piping for a Condensing Plant

In casting the pipe, if the core shifts slightly, or leans to one side, the pipe will be cast thicker on one side than the other. This is not easily detected in the finished pipe. The writer has seen cast-iron pipe broken, where the thickness of the pipe was intended to be 1½ inches all around. At the broken section, the thickness on one side was almost 2⅝ inches and on the other side only about ⅜ inch.

When ordinary brass pipe it should be distinctly specified to be "iron-pipe size" threaded to iron-pipe standard; otherwise the flanges and fittings may not match the pipe owing to the difference in outside dimensions.

The outside diameter of a given size pipe remains constant whether the pipe is standard, extra strong, or double extra strong, so that any grade of pipe may be screwed into any grade of fitting or flange of the same size.

Corrugated-copper gaskets are used between the flange faces in preference to rubber-composition gaskets, as they wear better and insure a tighter joint.

Suction Mains for Boiler-feed Pumps

The suction pipe for a boiler-feed pump should be absolutely tight and free from air pockets, and laid so that any air admitted through the intake would rise to the pump or suction air chamber and not be pocketed in some high part of the suction pipe.

To accomplish this result the suction pipe should gradually slope away from the pump. A slope of 3 inches to every 25 feet of pipe length, or 1 foot slope to every 100 feet of length is recommended by good authorities.

A very small air leak in the suction pipe may break the vacuum, caused by the pump plunger, and thus prevent the pump getting its proper supply of water, causing it to work improperly.

The end of the suction pipe should be equipped with a foot valve and strainer and placed from 6 to 8 feet under water (never less than 3 feet), to prevent air entering with the water.

In long suction mains a suction air chamber should be placed in the suction line, close to the pump cylinder to cushion the water, and for maintaining a uniform flow in the suction pipe, thus reducing the pump slip.

Feed-water Regulators

The water line in the boiler should be maintained at as nearly a constant level as possible. Keeping the water at a constant level in the boilers necessitates considerable attention on the part of the boiler attendants, especially with fluctuating loads. There are a number of feed-water regulators and pump governors which are designed to keep the water level constant.

In small plants, where the boiler attendant's duties are numerous, such devices are of considerable assistance. In large stations, however, engineers usually omit all such fixtures, as their action is not always positive, and on account of the trouble in caring for, and repairing them when out of order, preferring to rely on hand regulation. Where feed-water regulators, pump governors, etc., are installed they should be provided with a bypass connection for use when adjusting or repairing.

Economizers

The fuel economizer is a fuel saver, under certain conditions, but in a small plant where sufficient exhaust steam is available for heating the feed water they are entirely out of place.

Where the boilers are run at their normal rating, and the temperature of the flue gases is low, an economizer will reduce the temperature still lower and impair the draft by increasing the resistance of the gases passing from the boilers to the stacks, making it necessary to install a higher chimney, or install mechanical-draft apparatus to do the work of the natural draft. The economy resulting from the use of an economizer depends on the temperature at which the feed water enters and leaves the economizer, and also on the temperature of the escaping flue gases from the boiler furnaces.

Operating under the right conditions, there is no question but what an economizer would show a considerable saving of fuel, especially in large-size plants.

The Steam Turbine in Modern Practice*

A Brief Review of Basic Principles. Rapid Development of the Turbine to a Capacity of 14,000 Kilowatts. Possibilities of the Low-pressure Turbine

BY G. R. PARKER

The objective of designers of all classes of steam prime movers has been the same; namely, the conversion of the heat of combustion into mechanical or electrical energy; and the medium employed has been water. It is true that the over-all efficiency of our best steam prime movers is regrettably low, due to the fact that so much heat has to be given to water before any of it can be converted into mechanical work. For example, the total heat per pound of steam at 150 pounds gage pressure is about 1195 B.t.u. If steam be expanded to a 28-inch vacuum the total energy available in this range is only about 321 B.t.u. This 321 B.t.u. is all of the total heat we are able to use, and in practice commercial machines may convert anywhere from one-half to three-fourths of this available energy into mechanical work.

In this country we are interested chiefly in the Parsons and Curtis types of steam turbine. Briefly, the Parsons principle involves the continuous expansion of steam through alternate rows of moving and stationary blades, the former being attached to the spindle and revolving it, and the latter redirecting the steam against other revolving blades. The expansion of the steam thus occurs in both moving and stationary blades and the motion is given the rotating element, both by the impact of steam on the moving blades and its reaction on leaving them. The machine is ordinarily called a reaction turbine and may in a general way be compared with a reaction water wheel.

The Curtis principle differs from the Parsons in that the expansion, instead of being continuous throughout the machine, is broken up into a series of pressure steps or stages. Each one of these contains a row of stationary nozzles which expand the steam through a certain range and direct it with large velocity against the moving buckets, through which it passes with practically no further expansion, thus moving the revolving buckets by impulse only. The expansion of steam, therefore, occurs only in the stationary element. This type of turbine is referred to as the impulse type and is somewhat analogous to the impulse water wheel. It is with the impulse turbine as invented by Curtis and perfected by Emmet that this paper deals.

Most Efficient Speed

The problem confronting all turbine designers has been to reduce the speed to such an extent that the turbine itself and the generator connected with it could

*A paper read before the joint meeting of the American Society of Mechanical Engineers and the St. Louis Engineers Club.

be made to safely withstand the centrifugal strains. Other speed limits are imposed by the commercial electric frequencies in use in this country. Evidently 1500 revolutions per minute is the highest speed for a 25-cycle generator and 3600 revolutions per minute for 60 cycles.

The most efficient speed for any single impulse wheel driven by a moving liquid or gas is one-half that of the moving element. Steam exhausting from a pressure of 150 pounds gage through a suitable nozzle attains a velocity of about 4000 feet per second. Therefore, half this speed, or 2000 feet per second, should be the correct peripheral speed of a single impulse wheel placed in the path of the steam jet. Evidently such a peripheral velocity would necessitate an angular velocity far in excess of the highest commercial speeds. To reduce this high angular velocity and at the same time retain the efficiency of his machine, Curtis made use of two expedients. The first consisted in utilizing the velocity of a single expansion in more than one wheel, and thus dividing the initial velocity into two or more parts. This reduced the peripheral speed of each wheel in inverse proportion to the number of wheels. However, there are practical limits to the number of wheels which can be utilized in a single expansion; therefore, to still further reduce the speed, Curtis not only divided the velocity of a single expansion into two or more steps, but divided the total expansion range into two or more separate expansions.

Number of Stages

A considerable amount of experimenting was done in the early days of manufacture to determine the correct number of stages and wheels per stage. Thus the first large turbines built contained two stages and three rows of revolving buckets per stage. Later investigations showed that for all large turbines the most economical results were obtained with not more than two rows of buckets per stage, and this is the present standard. With the exception of very large machines, four stages have been regarded as the correct number, although recent experiments indicate that possibly greater economy may be obtained with one or more additional stages.

High Economy at All Loads

One of the principal and most justly founded claims for the steam turbine is its relatively high economy at other loads than rated load. Even in the best designed reciprocating engines the best economy is obtained at one point, and at loads greater or less than this, cutoff occurs either too early or too late for the cylinder proportions, and the result is a steam consumption per horsepower relatively higher at these loads. Since very few power loads can be made to hold constant at any given point, the average economy on a varying load may be considerably in excess of the best obtainable value.

On the other hand, the Curtis principle permits high economy at all loads. This is due largely to the method of governing. Most impulse wheels have partial peripheral admission of steam; i.e., steam flowing through only a portion of the wheel at one time. Thus in the Curtis type the nozzles expanding the steam and admitting it to the first-stage wheel extend over only a small portion of the wheel periphery. These nozzles are generally placed close together in a single continuous arc, although in the very large sizes two groups of nozzles spaced 180 degrees apart are employed. The admission of steam to these nozzles is controlled by a corresponding series of valves which vary in number according to the size of the machine. The opening or closing of these valves evidently permits the passage of steam through a greater or less number of nozzles. The steam emerging from two or more nozzles combines to form a continuous belt or stream of steam of constant width, determined by the width of the nozzles, and of length corresponding to the number of nozzles open. Governing is thus accomplished by automatically varying the length of this steam belt by the successive opening or closing of the admission nozzles. The steam thus arrives at the point of inlet at full pressure, regardless of the load, and whether one or all the nozzles are open it is expanded with practically no throttling. The result is evident in the steam-consumption curves. At fractional loads the steam consumption is relatively good, and as the load is increased the economy continues to improve, the load-water rate curve gradually becoming a nearly straight line. With such a machine it is possible to operate over at least half the range of the machine with maximum and minimum economy varying not more than 5 per cent. from the average. The advantages of this feature on a fluctuating load are obvious.

Vacuum, Superheat and Initial Pressure

The question is often asked as to what are the most economical steam conditions; i.e., initial pressure, vacuum and superheat. While there is much discussion on these points, the present American practice is becoming reasonably standardized. As to vacuum, there is no question that it is worth while getting the highest obtainable. Twenty-eight inches (properly speaking, 2 inches absolute) and even higher vacuum can readily be obtained with modern condensing apparatus. Steam pressures vary from 150 to 250 pounds gage. In the smaller and medium-sized plants probably 150 to 175 pounds is about right, while in the larger ones 175 to 250 pounds should be employed.

The arguments for and against superheat are numerous, but the consensus of opinion is inclined to favor a reasonable degree of superheat, at least in the large and medium-sized plants. Superheat ranging from 50 to 200 degrees Fahrenheit is in common use. Regarding high pressure and high superheat it should be borne in mind that the percentage increase of available energy given the steam is much greater than the percentage increase in fuel necessary to produce these conditions.

Rapid Development of Turbine

Modern steam-turbine practice has advanced so rapidly in the past few years that quite startling changes have been effected in some of the original turbine stations. Improvements in details of construction, and increase in speed, have greatly reduced the size and weight per kilowatt. About six years ago the first large turbines were installed in the new Fisk street station of the Chicago Edison Company. The first three machines were vertical two-stage machines of 5000-kilowatt capacity, and the fourth, installed somewhat later, was of the same capacity but of the five-stage type. Within the last year these four machines have been removed and replaced by four vertical machines of 12,000-kilowatt continuous capacity each. These occupy no greater space than the original machines, and no increase in the capacity of boilers supplying them was necessary. The Fisk street station now contains altogether ten similar machines of 12,000 kilowatts each. The Quarry street station of this same company at present contains three vertical machines of 14,000 kilowatts each and three more will be installed during the coming summer. The economies obtained in these plants are reflected in the rates which the Commonwealth Edison Company is able to make its consumers.

A somewhat similar evolution is now under way in St. Louis. In 1905 the present Union Electric Light and Power Company installed two 5000-kilowatt, 500 revolutions per minutes, 25-cycle, 6600-volt vertical turbines. Later, two more 5000-kilowatt machines were added, but with 60-cycle, 2300/4000-volt generators. The present plan, which is now well under way, is to replace all four machines with other turbines of 12,000-kilowatt capacity each.

The natural question is, can it possibly be a good business proposition to throw out four large turbines which have been in use only three or four years? That the answer was affirmative was due to three principal considerations.

(1) The larger machines could be installed without increase in floor space.

(2) The improvement in economy rep-

resented an annual charge which, if capitalized, would more than pay for the additional investment.

(3) Practically no new auxiliary apparatus or station piping would be required.

That these considerations were based on correct assumptions has been amply demonstrated. The first 12,000-kilowatt turbine has now been in commercial operation for several weeks.

In making this installation not only was it possible to utilize the original foundations, but even the base of the old turbine, which also constitutes the exhaust chamber and step-bearing support, was utilized in building the larger machine. It was, therefore, not necessary to remove this base from the concrete, or break the connection to the condenser. In a general way it may be said that the old 5000-kilowatt turbine was lifted bodily from its exhaust base and a 12,000-kilowatt machine installed in its place, without disturbing the condenser piping and auxiliaries. The increase in capacity means that in this portion of the station the kilowatt per square foot of station has been more than doubled. Even the original four 5000-kilowatt machines were placed unusually close together, and when the remaining three are replaced by larger machines, it seems probable that the turbine portion of the Union Electric Light and Power Company station will show a greater kilowatt capacity per square foot than any other station in this country.

Aside from the increase in capacity, the improvement in steam economy is very large. Unfortunately, detailed test figures are not available at this time, but it is probable that the new turbine will show an improvement of at least 20 per cent. over the one which it replaced, besides having a flatter load curve. On this basis considerations of economy alone would have warranted the change. In addition, this enormous increase in power has been effected without any considerable change in the existing piping and auxiliary arrangements.

Any steam turbine operating condensing with the usual pressure and vacuum derives roughly half of its power from the expansion of steam from boiler pressure to atmosphere, and the other half from the remaining expansion from atmosphere to vacuum. Thus in a four-stage machine, atmospheric pressure is reached between the second and third stages. A reciprocating engine operating under similar conditions would not derive its energy from the steam in this proportion. The average engine actually develops some 15 to 20 per cent. of its power from the expansion of steam below atmosphere.

Low-Pressure Turbine

A consideration of these facts brings us to the low-pressure turbine. In a general way, it may be stated that a low-pressure turbine is that part of the high-pressure turbine which normally operates below the atmospheric line. In general, it is possible to build a low-pressure turbine for compounding with a noncondensing engine, with the expectation of approximately doubling the capacity and halving the steam consumption, while the same thing may be done with a condensing engine to a less extent. Most condensing engines can be operated at their full capacity noncondensing with a slight adjustment of the valve gear. The increase in steam consumption of the engine alone with this arrangement will be between 15 and 25 per cent.; but the low-pressure turbine adds 90 to 100 per cent. capacity, so that the net economy effected is worth going after regardless of the tremendous increase in capacity. Installations of this character were first made in this country four or five years ago, although greater interest has been stimulated during the last year or two. One of the early installations was in the plant of the East St. Louis and Suburban Railway Company, where an 800-kilowatt and a 1000-kilowatt low-pressure turbine were installed in connection with noncondensing engines.

The most notable installation is that recently made in the power house of the Interboro Rapid Transit Company in New York City. This station contains the highest type of reciprocating engines, operating under the best possible engine conditions and developing an economy comparable with the best engine station in the country. It has, however, been possible to install in connection with one of these engines a low-pressure turbine with a nominal rating of 5000 kilowatts. The improvement in steam consumption of the combined engine and turbine has effected a saving in coal consumption of over 20 per cent. and the combined capacity has been more than doubled. The turbine generator is of the induction type and runs permanently in parallel with the engine generator. With this arrangement it is unnecessary to provide any speed governor on the turbine. The engine governor takes care of both machines. It is interesting to know that in spite of the size and special character of this installation the machine was started and placed in commercial service without a hitch of any kind. A second machine of similar characteristics, but with a larger generator, is now being installed in connection with the second engine, and the present plan contemplates one turbine for each engine throughout the station. When complete, the station capacity will have been more than doubled without any increase in real-estate or building investment.

What has been done in this connection can be accomplished on a smaller scale in almost any plant of 300 kilowatts or larger, operating reciprocating engines either condensing or noncondensing, provided proper condensing facilities are available.

A valuable feature in connection with the Curtis low-pressure turbine is that, owing to the fact that even with low-pressure steam the primary admission nozzles only extend a portion of the way around the wheel circumference, it is possible to equip any low-pressure machine with another set of nozzles primarily designed to expand steam from boiler pressure instead of from atmospheric pressure. A machine so equipped can be operated either as a strictly low-pressure machine, or should the supply of exhaust steam fail entirely due to shutdown of the engine or other reason, it can operate and carry its full capacity on boiler-pressure steam alone; or it can be operated on a mixture of the two, in case the load exceeds the supply of exhaust steam. It should be remembered that these high-pressure nozzles do not throttle the steam to a lower pressure, but are actually designed to economically expand it to the proper internal pressure of the turbine. Such a machine operating on high-pressure steam only, will show an economy fairly comparable with an engine or turbine regularly designed for high-pressure operation. The operation of these high-pressure nozzles is automatically controlled by the main governor, and in practice it has been found possible to instantly cut off the low-pressure steam supply without a noticeable variation in the speed of the turbine. This evidently makes a most flexible machine and one that accomplishes two most desirable results at comparatively small cost, namely, increase in capacity and decrease in steam consumption.

It is also perhaps worth while mentioning the enormous field which has been opened up by the strictly small turbine, that is, from 300 kilowatts down. These machines for the most part are designed to operate noncondensing, and the argument in their favor is that they are extremely simple machines requiring practically no attention or adjustment. The best proof of their extremely rugged construction is found in the fact that, out of 500 small turbines of 25- and 35-kilowatt capacity now operating in various parts of the country, a large percentage are in use by the various railroads for electric train lighting, under which condition it is hardly necessary to say that they receive a minimum of attendance with very few opportunities for the making of repairs. Machines of this size are also in general use as exciters.

In conclusion it may be said that the Curtis turbine has been built and placed in successful operation in sizes from 5 to 14,000 kilowatts, and at the present time even larger machines are under consideration.

ELECTRICAL DEPARTMENT

ESPECIALLY CONDUCTED *to* BE *of* INTEREST *and* SERVICE *to the* MEN *in* CHARGE *of the* ELECTRICAL EQUIPMENT

Lessons of Some Motor and Dynamo Failures

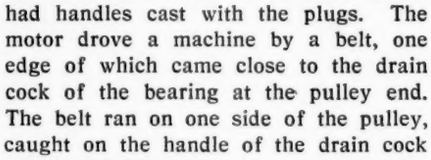
By Howard S. Knowlton

The study of machinery failures is always of interest to the operating and designing engineer, for it is only through analysis of the defects and inefficiencies of equipment in actual service that improvements leading to more economical operation and more reliable apparatus can be effected. The failures of electrical machinery are particularly instructive when the detailed conditions of the occurrences are known, for in no other class of apparatus are apparently small and trifling causes so prolific of large results. In the following paragraphs an outline of a number of breakdowns occurring during the past year in motors and generators is presented, based upon extended reports of the engineering expert of a large accident insurance company making a specialty of underwriting power plant machinery.

The case occurred of a heavy flashing in a two-pole direct-current generator direct driven by a compound vertical engine at 500 revolutions per minute; the machine voltage was 110. On starting the engine after it had been overhauled by the makers a serious flashing occurred at the dynamo armature. When the machine was examined four of the armature conductors were found to be nearly cut in two; the end connections of six others were bent and the insulation was torn off all around the armature at the end nearest the engine. The cause of the trouble was a small tapered pin which had been laid down near the dynamo and had become drawn into the air gap when the field-magnet was excited. The cost of repairing the damage was all out of proportion to the simplicity of the careless act which led to the trouble. A somewhat similar case was that of a two-pole semi-inclosed shunt-wound 2-horsepower motor driving a nailing machine. One of the nails fell into the casing and was drawn into the airgap, short-circuiting some of the armature conductors. The armature had to be entirely rewound on account of the location of the damaged coils. Both of these cases illustrate the danger of leaving small pieces of iron or steel near generators or motors, especially those of the open type, with stray fields unscreened by iron or steel casings.

A breakdown occurred in a direct-current 5-horsepower motor, running at 700 revolutions per minute. The armature shaft was carried in phosphor-bronze bearings lubricated by rings running in oil wells fitted with drain cocks which had handles cast with the plugs. The motor drove a machine by a belt, one edge of which came close to the drain cock of the bearing at the pulley end. The belt ran on one side of the pulley, caught on the handle of the drain cock and knocked it open; the result was that the oil ran out of the well, lubrication ceased and the bearing seized. The lesson of the accident was obviously that drain cocks should be made in such a way that they cannot readily be opened by the chance touch of a jacket, a cleaning rag or belt, if one be in the vicinity.

Lack of cleanliness in lubrication is responsible for no little trouble in the operation of electrical machinery. A case of this class occurred with a 5-horsepower motor running at 960 revolutions per minute, which blew its fuse persistently. When examined it was found that owing to the melting of the white metal in the bearing next to the pulley, the armature had been rubbing against the poleface. One of the wire binders had been stripped off, cutting up the insulation of the conductors. The oil ring of the damaged bearing was so full of dirt that the ring could not revolve and consequently the journal ceased to be lubricated and the white metal melted. Long neglect of the conditions was clearly responsible for the trouble. In another similar case four of the armature conductors were fused underneath the binding wires next the commutator and three other bands were nearly worn through by rubbing on the polefaces. The cause of the trouble was excessive wear of the gun-metal bearings which carried the armature shaft, clearly due to neglected lubrication.



Dovetail Brush and Holder

Oil getting behind the commutator lugs of a compound-wound 100-volt generator was the cause of a serious breakdown. The commutator was grounded, the mica insulation between bars burned, the conical clamping ring at the back of the commutator fused, and the solder holding the armature conductors to two of the cummutator lugs was melted. The cause was short-circuits between four of the armature bars and the clamping ring at the end next to the armature. The short-circuits were established by an accumulation of oil and carbon dust behind the commutator lugs. The owners of the machine had been frequently advised of the importance of keeping this vulnerable part of the machine clean, but, while an air compressor was used for the purpose, the presence of the oil rendered the blast ineffective. The commutator was taken off and the damaged insulation and clamping rings were replaced. The lugs were resoldered to the armature conductors. The report states that passage of oil to the rear ends of commutators is very difficult to prevent and most troublesome in its effects, as it cannot be removed. It does not take place when commutators are pressed upon their shafts, but no care in fitting seems able entirely to prevent it. When quills are held by feathers or screws the only satisfactory way appears to be to calk a copper ring between the shaft and the front end of the casting.

Improper location of a motor resulted in the following accident. The machine was a semi-inclosed induction motor and was located in a chamber at the top of a tower, into which hot ashes slacked with water were lifted by an endless chain driven by a belt. The chamber was full of dust and steam. The insulation of the stator windings became so decayed through the action of the sulphurous steam and impregnated with grit and dust that they became short-circuited. The fuse blew and four of the stator coils were burned, the top and bottom in one slot and the top in two adjacent slots. The rotor suffered no visible damage. The motor was moved to a sheltered place on wall brackets and no further trouble occurred.

In the case of a four-pole shunt-wound motor of 6 horsepower, wound for 220 volts, three of the commutator segments were short-circuited by copper dust and partly fused, and the insulation was burned off three of the armature coils. Examination of the motor showed specks of copper imbedded in the mica between the bars and some of the edges burred over. Inquiry developed the fact that the owners of the machine had been turn-

ing the commutator in their own lathe, and had started it in service without taking care to clean the copper dust out of the mica between the bars or to prevent bridging from bar to bar before putting the armature back in the machine. The case was typical of many similar failures. After commutators have been turned the most careful examination is essential, the weakest point being in the corners where the lugs rise from the bars.

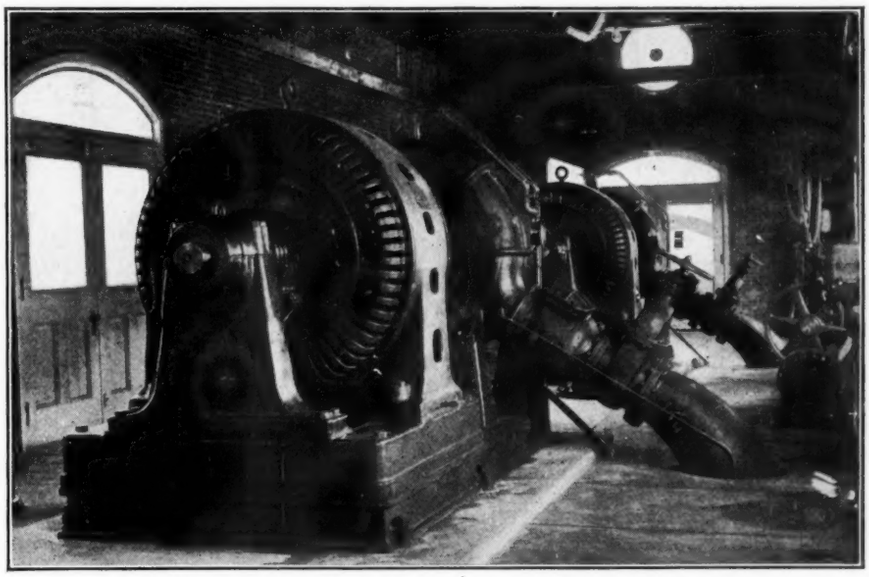
A motor of the inclosed type got into trouble through poor brush-holder arrangements. Each of four holders contained one carbon brush held in a dovetail slot through the end of the brush holder, as in the photograph. The brush was prevented from coming out of its slot by a small set screw passing through the top of the holder and pressing on the flat top of the brush. Upon complaint that something had gone wrong with the motor an inspector was sent to examine it, and he found one of the brushes in the bottom of the motor frame and its holder resting on the commutator and considerably worn down. It was evident that the brush must have fallen out at an earlier date and allowed the holder to drop upon the commutator, there being no stop upon the spindle to prevent this. In this case the holder itself was unsatisfactory; otherwise it would not have allowed the brush to fall out. The insurance company has had the least trouble with brush holders of the simple box type holding a rectangular block of carbon, with a flexible conductor attached and a spring to press the carbon against the commutator surface.

The results of a slack fit were shown in the instance of a 10-horsepower motor of the semi-inclosed type, which was wound for a normal speed of 800 revolutions per minute It was used to drive a pump through gearing. Owing to the fusion of two of the armature coils in their slots the motor was removed from service for repairs. Several of the conductors were found to be broken back of the commutator spokes, and both the commutator and the armature were so loose on the shaft that they could be moved by hand. When the commutator was taken off the nut which held the core plates was found screwed back off the thread on the shaft, so that all the plates were loose. Forty-six out of a total of 88 conductors were broken and the rest were in bad condition. The nut had been a slack fit on the shaft and had not been fastened by lock nut or taper pin.

A representative accident occurred with a four-pole 12-horsepower shunt-wound motor making 1200 revolutions per minute at 250 volts. The fuse blew and the inspector found the armature windings so short-circuited and grounded that the armature had to be stripped to get at the damaged coils. There were six of them, and all six had been in contact with the supporting drums at the ends of the armature, the sharp unprotected edges of which had cut through the insulation. The accident suggests, according to the report, that the edges of end plates and extension drums over which thinly covered conductors pass should be well rounded, and that the coils should be suitably protected by insulating material. It is not possible absolutely to prevent relative movement between the conductors and the core or drum, and it is certain that any movement, however slight, will enable a sharp metal edge to cut through the thin covering of the armature coils. This relative motion is also responsible for accidents in which armature coils are severed from their junctions with the commutator lugs. Ordinary methods of preventing the rotation of armature spiders and commutator quills, and still more, of armature core plates threaded upon the shaft, by means of feather keys, screw heads and set screws, are not effective, either because the bearing surfaces are insufficient or because the fitting is not good enough. The best plan when the diameter permits is undoubtedly to build both armature and commutator upon a common central casting or spider.

Motor-Driven Pumps at Lockport, N. Y.

The city of Lockport, N. Y., derives its water supply from the Niagara river near North Tonawanda, thirteen miles away. From the electrically driven pumping station at this point, the water is forced through 69,000 feet of 30-inch main to a standpipe 25 feet in diameter and 120 feet high, at Lockport.



TWO OF THE LOCKPORT MOTORS AND PUMPS

The pumping plant contains three independent units with a combined capacity of delivering 15,000,000 gallons in 24 hours against a pressure of 125 pounds. Each unit comprises a Westinghouse three-phase induction motor, designed to develop 500 horsepower at 750 revolutions per minute, direct-connected to a single-suction inclosed-impeller centrifugal pump made by the Holly Manufacturing Company, Buffalo. The power for the motors comes from one of the big generating plants at Niagara Falls, and is delivered to North Tonawanda at 22,000 volts; it is stepped down to 440 volts at the pumping station by three 500-kilowatt oil-insulated transformers, of the water-cooled type.

RESULTS OF TESTS OF MOTOR-DRIVEN PUMPING UNITS.

	Pump 1.	Pump 2.	Pump 3.
Average revolutions per minute	743	744	743
Average gage pressure, pounds	136.9	142.6	138.9
Average equivalent head, feet	315.8	329.1	320.6
Gallons pumped per hour	238,000	236,300	236,750
Average kilowatts input	342.4	346.1	341.5
Hydraulic horsepower	315	326	318
Electric horsepower	459	464	458
Efficiency, per cent	68.6	70.3	69.4
Degrees temperature rise in 15 hours	41	38	43

The three motor-driven pumping units were recently given a very complete test, which demonstrated the economy of electric operation. Each unit was run continuously for 15 hours, during which time readings of the electrical input were taken every five minutes; readings of the Venturi water meter every ten minutes; pressure gage every five minutes, and the speed of the pump every fifteen minutes. The results are given in the accompanying table, which shows that all three of the pumps exceeded the duty

requirements of 27,000,000 foot-pounds by approximately 4,000,000 foot-pounds. It will be further noted that the temperature rises are well within the specifications, and that the pump capacities and horsepowers are well exceeded. The electric power for the motors costs $16 per horsepower per year.

In connection with this plant, which is pumping its output about 13 miles to Lockport, it may be of interest to note that last summer when a large fire broke out there, eleven powerful fire streams through varying lengths of hose were maintained from the standpipe and pumps as long as needed for service.

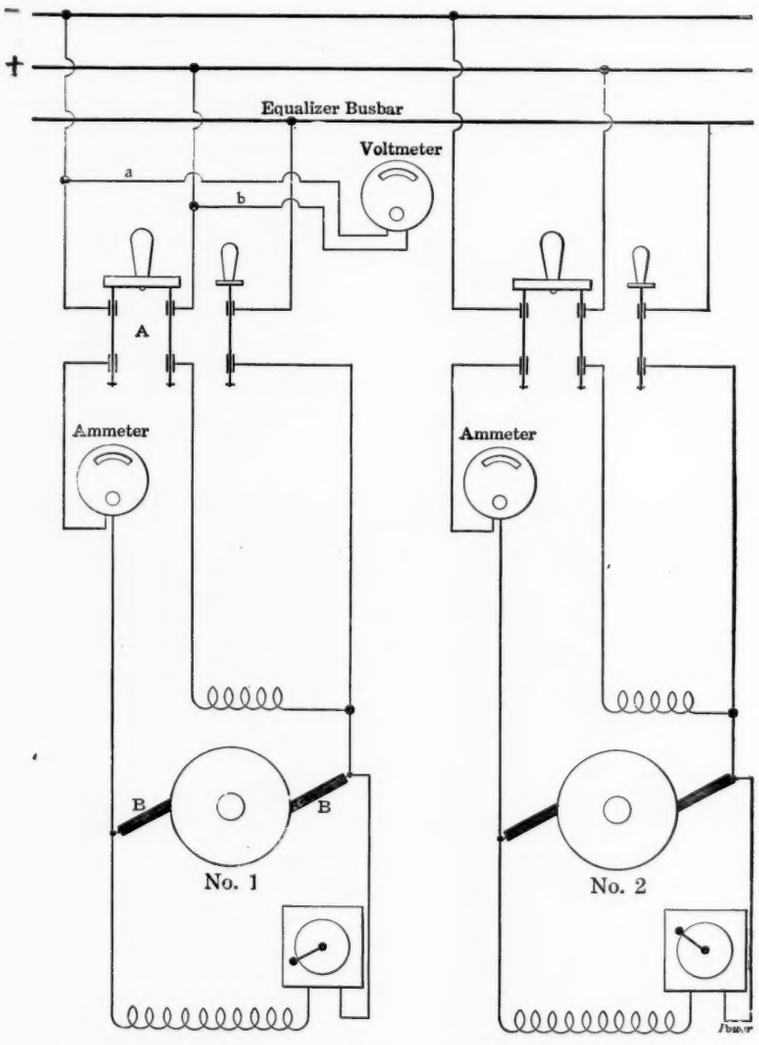
CORRESPONDENCE

Correcting Reversed Polarity

In the article on correcting reversed polarity, by A. W. Griswold, in the issue of February 1, are some statements that might create a wrong impression and on this account criticism will not be out of place.

In the first place, with a voltmeter of the permanent-magnet type there is no excuse for paralleling two machines of opposed polarity, as the voltmeter will correctly indicate the polarity of each and show at once if anything is wrong. Some voltmeters will not indicate polarity but register only the e.m.f. of the circuit. Of this kind may be mentioned the hot-wire instruments and those in which the pointer is actuated by the reaction between a stationary coil and a pivoted coil.

Reversed polarity may be indicated by the ordinary Weston ammeter common to nearly all switchboards. If the load can be thrown on the machine separately the ammeter will read normally if the polarity is correct and try to indicate in the opposite direction if the machine is reversed. It was not necessary for Mr. Griswold's chief to start up the No. 1 machine in order to get the polarity right. The process he describes is that of righting the polarity with the shunt-field winding and might have been done as well with the armature standing still.



MR. GRISWOLD'S WIRING DIAGRAM OF THE GENERATOR CONNECTIONS

From reading the description one would conclude that it was No. 2 machine that was reversed and after the process of "correcting" No. 1 both were really reversed. He says, "then he put the brushes back in the holders of No. 1 and turned the rheostat around until the pilot light lit up but the voltmeter did not register." If the voltmeter was properly connected in the first place this of itself would indicate that the polarity was wrong in both machines, since the process described would give both machines the same polarity. If the voltmeter was properly connected the fact that the leads *a* and *b* had to be transposed to make the instrument register proves that the polarity of the machine was wrong and since the two operated in parallel it follows that both were finally reversed.

A good way to right the polarity of a reversed generator is as follows, referring to Mr. Griswold's sketch. Suppose that No. 2 is right and No. 1 is reversed. Now to get No. 1 right it is only necessary to put the load, or a portion of it, on No. 2 and close both equalizer switches and the switch in the lead from the series coil of No. 1. The armature current from No. 2 will then divide between the series field windings of the two machines, and the magnetizing effect will right the polarity of No. 1, since the current flows through the series coils in the same direction that it would if No. 1 were delivering current of correct polarity to the busbars.

This method does not involve using the shunt field winding, with its resultant high e.m.f. induced when the field circuit is broken, as in the method described by Mr. Griswold. He is also mistaken as to the cause of this high e.m.f. He says the long arc at the switch may set up a high voltage in the field winding. The arc at the switch is not the *cause* of the voltage but the *result* of it.

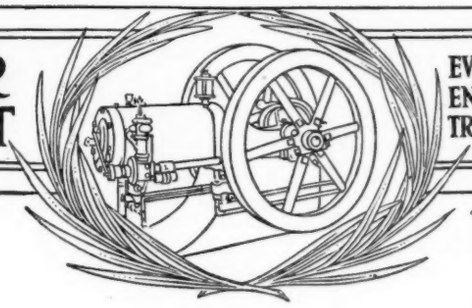
The cause is the "collapse" of the field magnetism when the circuit is broken; any change in the magnetism of an iron core surrounded by a winding will induce an e.m.f. in that winding, proportional to the number of turns in the coil and the rate of change in the magnetic lines of force in the core. There are a great many turns in the shunt field coils, and when the circuit is opened the magnetism dies out almost instantly, so that the "rate of change" is also very high.

There is no good reason for taking the load off of No. 2 before pulling the switch on No. 1, since pulling the switch on No. 2 has precisely the same effect on No. 1 as though the switch *A* had been opened. Opening either switch interrupts the shunt-field current of No. 1. The only way to avoid the flash and consequent induced voltage is to open the switch *very slowly* or to let the voltage of No. 2 die down by either cutting in the field resistance or shutting down the engine before opening the main switch.

C. L. GREER.

Handley, Tex.

GAS POWER DEPARTMENT



EVERY THING WORTH WHILE *in the* GAS ENGINE *and* PRODUCER INDUSTRY WILL BE TREATED HERE *for the* USE *of* PRACTICAL MEN

The Gas Turbine

By A. W. H. Griepe

It is generally recognized that one of the chief difficulties in the way of devising a practical gas turbine is to find a material for the vanes which will be capable of withstanding the effect of being in contact with gases at the high temperatures produced by combustion.

PROPERTIES OF GASOLENE AND AIR.

Test Number	Mixture Volume. Gasoline Vapor.	Mixture Volume. Air.	Time in Seconds between Ignition and Highest Pressure.	Explosive Force in Pounds per Square Inch. Compression of Air in Pounds per Square Inch. 42	57	71	Temperature of Combustion* in Degrees Fah. Actual.	Temperature of Combustion in Degrees Fah. Theoretical.	Explosive Force in Atmospheres. Compression of Air in Atmospheres. 3	4	5	Temperature of Combustion* in Degrees Celsius. Actual.	Temperature of Combustion in Degrees Celsius. Theoretical.
1	1	13	0.28	156	208	260	1,857	3,542	11.0	14.6	18.2	1,010	1,950
2	1	11	0.18	183	244	305	2,196	4,010	12.85	17.15	21.4	1,200	2,220
3	1	9	0.13	234	312	390	2,803	4,806	16.4	21.9	27.38	1,535	2,650
4	1	7	0.07	261	348	435	3,119	6,001	18.3	24.4	30.5	1,715	3,325
5	1	5	0.05	270	360	450	3,226	6,854	18.95	25.35	31.6	1,772	3,790
6	1	4	0.07	240	320	400	2,965	5,517	16.85	22.45	28.1	1,625	3,040

*At atmospheric pressure.

The actual temperature of the gases, however, though high, does not approach the theoretical temperature, as is shown in the table, the figures in which they obtained in laboratory tests made by Dr. Richard Wegner, of Heidelberg, Germany.

Forcing steam or water into the rotor to cool the blades has been suggested as a means of preventing their destruction by the high-temperature gases. This arrangement, to be effective, would no doubt complicate the construction of the turbine considerably and would probably reduce its efficiency.

An excess of air in the mixture has also been suggested as a means to this end. The effect on the explosive force of the mixture is clearly shown in the table. Several attempts have been made to use blades made of heat-resisting materials, such as quartz and slate, etc. Hard porcelain reinforced by wire mesh has also been tried, but with very little success.

Blades made of vanadium steel and covered with a layer of magnesia have shown the best results so far but have not proven sufficiently good to warrant the use of this combination.

Fig. 1 represents diagrammatically a gas turbine of the velocity-stage type coupled to a three-stage centrifugal air compressor, as proposed by Dr. Stolze, of Charlottenburg, Germany, and described in a paper read several years ago. The gas, which is controlled by the flyball governor, enters the explosion chamber under pressure and is mixed there with air (under pressure from a three-stage centrifugal air compressor.) The mixture is fired in the chamber after which the air and hot gases, expanding in the nozzle, enter the turbine, where they give up their energy in much the same manner that steam does in the steam turbine, passing the rotating and stationary blades on their way to the exhaust outlet.

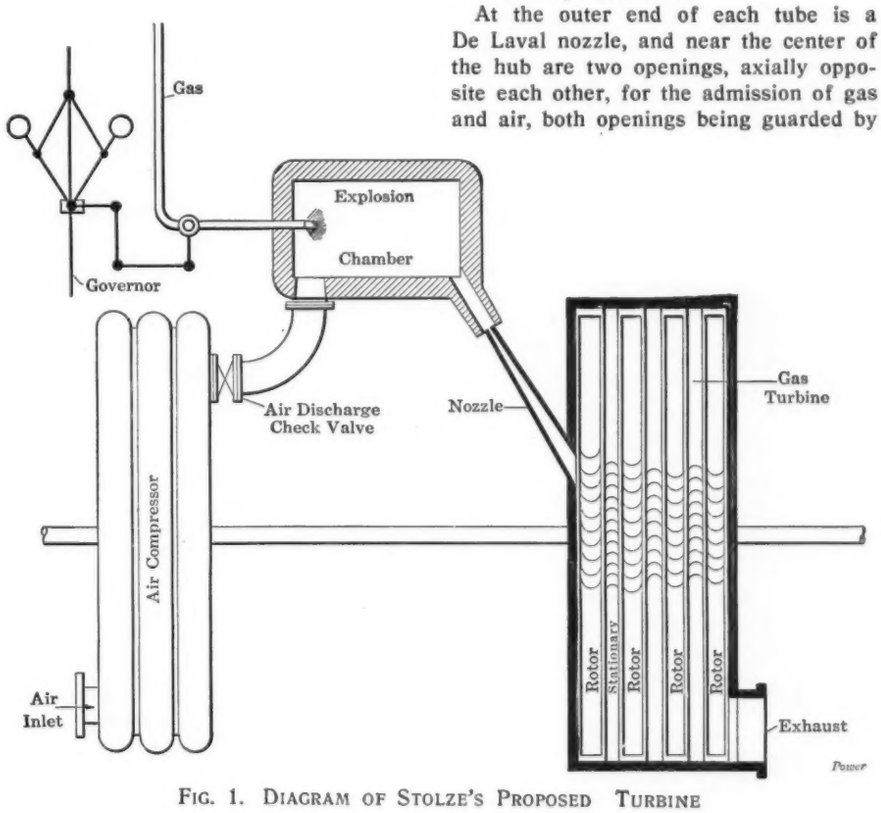


Fig. 1. Diagram of Stolze's Proposed Turbine

An improvement has been proposed for this type of turbine unit which consists in forcing steam or water into the combustion or explosion chamber in order to reduce the temperature of the burning gases to a temperature that will be safe for the vanes.

Fig. 2 is a more or less diagrammatic sketch showing an elevation and cross-section through a reaction type of turbine proposed by Doctor Wegner and described in a paper written two years ago.

The turbine rotor is made up of four tubes, closed at the outer ends and flanked on each side by round plates, as indicated. These plates are provided with circular openings which admit air for cooling the tubes after each explosion. (See the elevation.) The tubes and plates revolve on a shaft in the direction indicated by the arrow.

At the outer end of each tube is a De Laval nozzle, and near the center of the hub are two openings, axially opposite each other, for the admission of gas and air, both openings being guarded by check valves. The gas check valve, as will be noted, is connected to a ring which passes in front of a segment with a circular opening which admits the gas.

A spark plug is attached near the outer end of each tube, as shown in both views. When passing by the gas-admission segment, each tube is filled with gas and air through the check-valve ports. The gas is under a higher pressure than the air and therefore forces the air check valve to its seat, and as the turbine revolves further the centrifugal force compresses the mixture in the outer ends of the tubes. The spark plugs then ignite the mixture, after which the hot gases, escaping under pressure through the De Laval nozzles, force the turbine to revolve in the direction opposite to the outflow of the gases.

After the explosion occurs the check valves open again, admitting cool air which clears the tube of the burnt gases, at the same time cooling the tubes. As the casing and tubes revolve, air is sucked in through the cooling air inlets (shown in the elevation) which further facilitates the cooling of the tubes after each explosion.

Four explosions occur for each revolution of the turbine, and the reaction of the outflowing gases against the atmosphere is the source of the power. The turbine is therefore a reaction turbine in its simplest form, making separate air compression unnecessary.

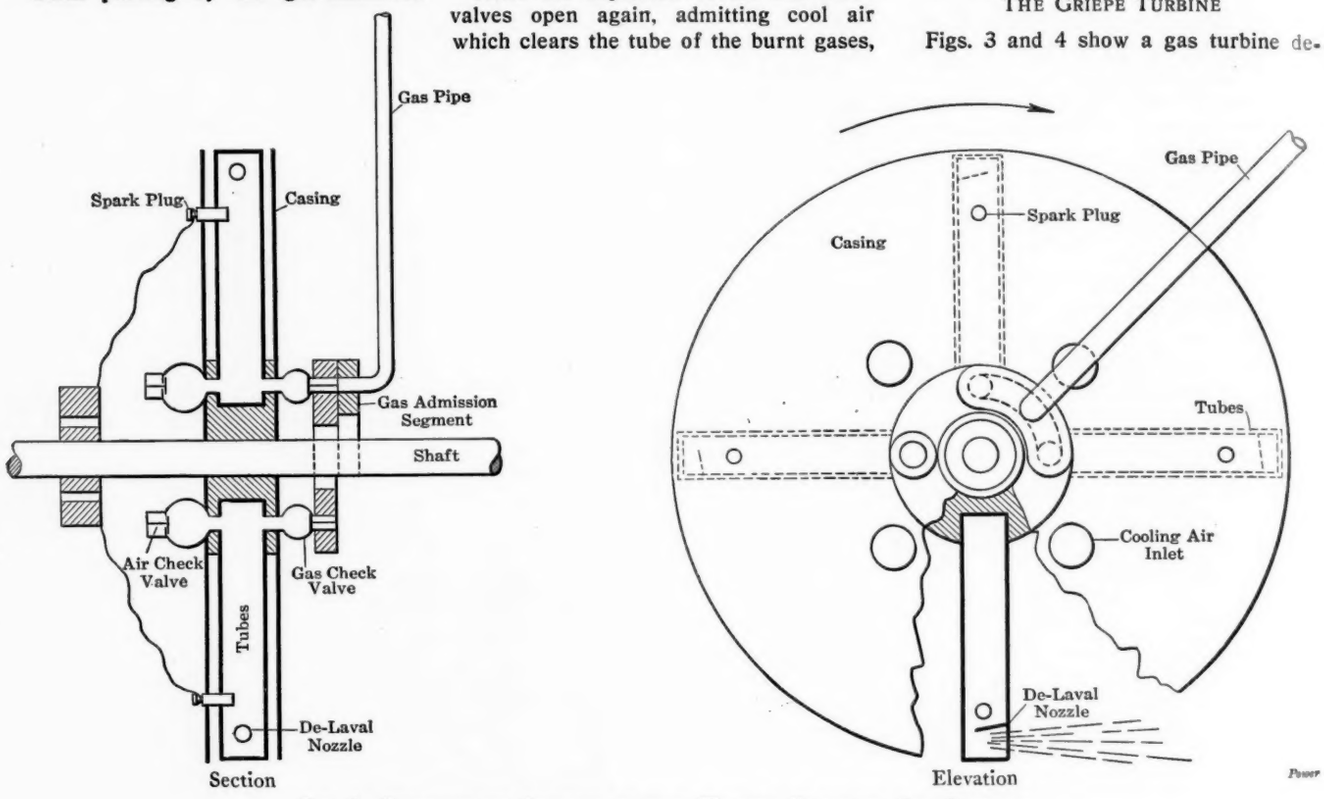


FIG. 2. ELEMENTARY SKETCH OF THE WEGNER PROPOSED GAS TURBINE

THE GRIEPE TURBINE

Figs. 3 and 4 show a gas turbine designed by the writer, which is really a combination of the gas engine and gas turbine. The unit consists of the turbine, a three-stage centrifugal air compressor, an air accumulator and a gas

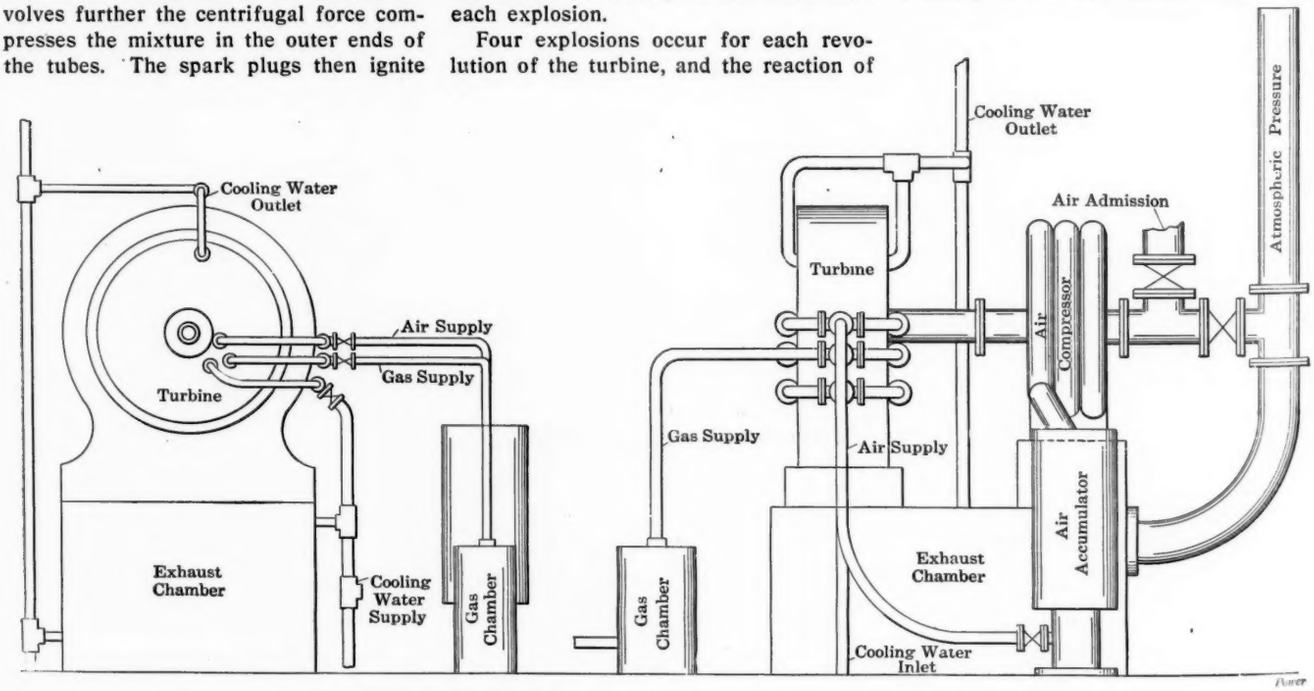


FIG. 3. COMPLETE ARRANGEMENT OF GRIEPE'S PROPOSED GAS-TURBINE UNIT

chamber. The whole unit is set on a common base which is built out as an exhaust chamber.

As will be noted from the following description, the flow of hot gases against the blades is intermittent, which gives the blades time to cool somewhat between the moments when they are in contact with the gases.

The rotor *A*, Fig. 4, which is shown in black, is made up of a series of expansion chambers containing blades *B* and a series of closed compartments *C* which separate the expansion chambers. The rotor *A* carries a ring *D* having openings *E* to admit gas from the gas chambers to the explosion or combustion chambers *F*.

Contact pieces *G*, fastened to the rotor, close the electric circuits of the spark plugs, which are located in the walls of the explosion chambers. Explosion chambers are provided on both sides of the rotor web, but are staggered angularly to avoid dead positions.

The main structure contains compartments for compressed air, cooling water, and gas, as shown.

The explosion chambers *F* are located on the outer part of the central structure and are alternately closed and opened to the expansion chambers by the inner faces of the compartments *C*. The explosion chambers may be as shown, or carried completely around the circumference of the central structure, according to the size and power of the turbine.

Compressed air from the compressor passes through the four-way valves *K*, which act as injectors. The air draws a certain quantity of gas along with it while passing through the valves, and the mixture is forced into the combustion chambers whenever the admission openings *E* in the rotor ring *D* register with the channels to the combustion chambers. After these openings have passed, closing the channel again, the revolving contacts *G* close the spark-plug circuits and the mixture is exploded. During this part of the revolution of the rotor, the blade compartments are nearly opposite the explosion chambers, and an instant later they register and the gases escape through the vanes, forcing the rotor in the direction of the arrow. Further motion of the rotor brings the compartments *C* again over the openings of the combustion chambers, and the cycle is repeated.

In the design shown here it is evident that the vanes or blades *B* are not continuously in contact with the high-temperature gases, as during three-quarters of a revolution of the rotor the blades are surrounded by gases at a lower temperature cooled by air admitted through the air channels *L* into the exhaust chamber.

The centrifugal air compressor driven by the turbine discharges into an accumulator to insure uniform air pressure. The density of the explosive mixture of gas and air can be regulated by adjusting the air pressure.

A further change in the output of the turbine can be attained by opening or closing one or more of the four-way valves *K*, thus changing the number of explosion chambers in action.

Consul Albert Halstead reports as follows concerning the increased use of electrical motive power in Birmingham since that English municipality bought out the Birmingham Electrical Supply Company in 1901:

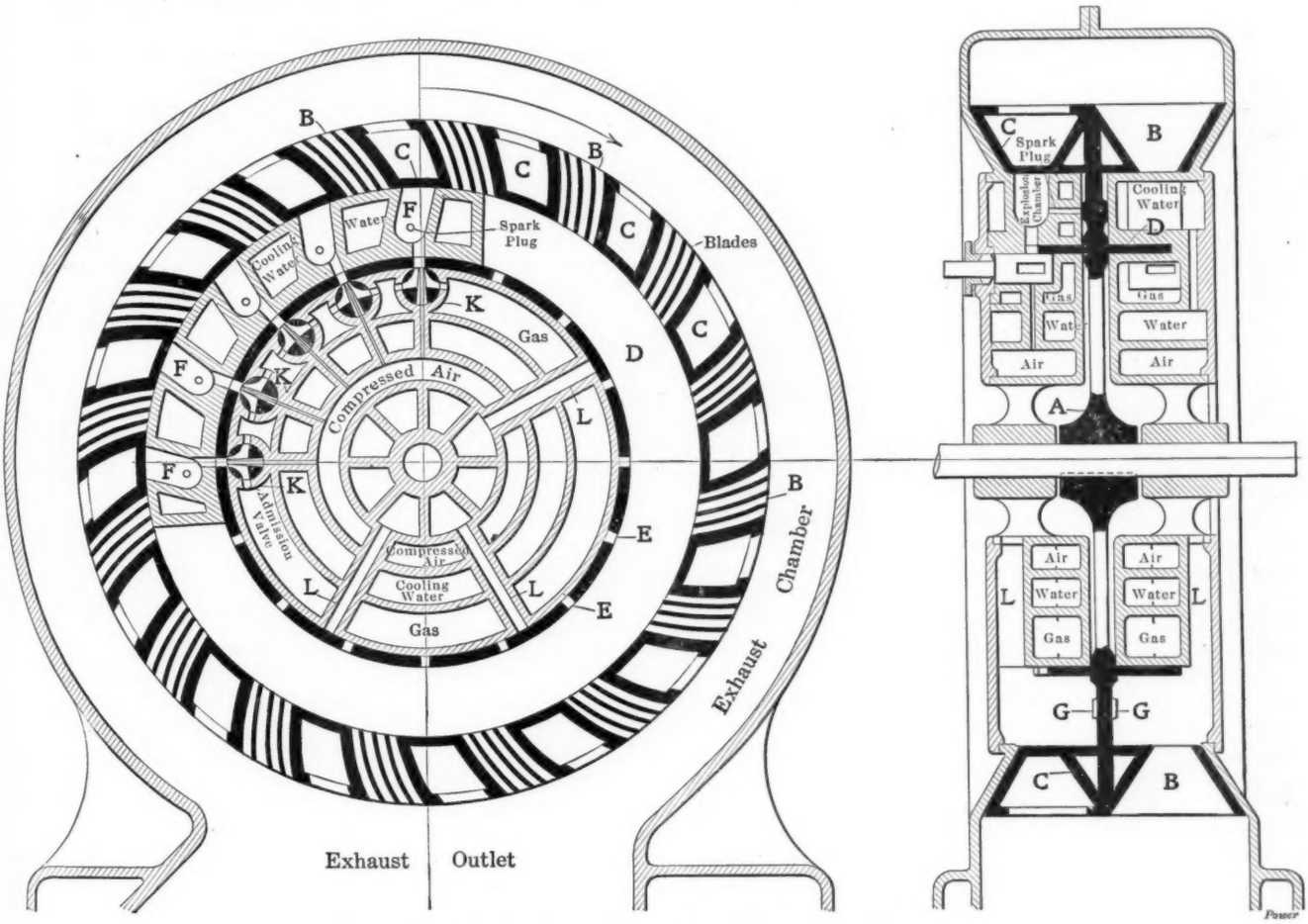


FIG. 4. VERTICAL SECTIONS OF THE GRIEPE TURBINE PROPER

In the year ended March 31, 1901, the total units sold for lighting and power purposes amounted to 3,040,822, while for the year ended March 31, 1909, the units aggregated 12,623,818. This does not include 11,782,409 units sold to the city tramways for traction purposes. Including these the total for the year ended March 31, 1909, was 24,406,227 units, or eight times the amount supplied when the city undertook control. A particular point has been made of pushing the sale of electricity for motive purposes, with good results. In 1901, 750 horsepower of electric motors were connected with the city supply, and in 1909 this had grown to 14,365 horsepower.

FROM READERS *with* SOMETHING TO SAY

PRACTICAL INFORMATION *from the* MAN *on the* JOB: A LETTER GOOD ENOUGH *to* PRINT *is* GOOD ENOUGH *to* PAY *for*

Unusual Cause of Scored Cylinder

Some time ago while inspecting the cylinder of our 18x48-inch Harris Corliss engine, I found the sides of the cylinder walls to be slightly scored. An inspection of the piston and rings failed to reveal the cause of the scratches. The engine was put together again and kept running; but the scratches worried me and caused me to take the head off frequently for the purpose of examining the cylinder, which I found to be getting continually worse. What seemed strange, and puzzled me the most, was the fact that the scratches were all confined to the sides of the cylinder while the bottom and top were free from marks and in as good condition as when the engine was installed. The cause of the scratches was no more apparent than in the first place, and as the cylinder had commenced to leak considerably, I decided to bore it out and make a new set of packing rings in the hope that by so doing I would remove the cause of the trouble, which to me was a hidden mystery, at the same time that I stopped the leakage of steam.

The engine had run about a week after being bored out when I again took the head off and to my amazement I saw new fine scratches again on the side walls of the cylinder. I was inclined to blame the to oil now. The head had been on but a couple of days when the gasket commenced to blow out a little, so that night I took the head off for the purpose of putting on a new gasket and while cleaning the surface from particles of old packing, I discovered the cause of the cylinder cutting.

When the head was fitted to the cylinder the nipple on the head was made a trifle longer than it should have been so that it pushed in past the steam port about ¼ inch, or, in other words, the chamfer or bevel on the top side of the nipple, made for the purpose of wedging the steam as it enters the cylinder, was not cut back as far as it should have been, consequently the steam coming into the cylinder had worn a V-shaped groove on the top side of the nipple. After years of constant wear the edge of the groove was brought to a knife edge, and then little particles began to crumble and fall off into the cylinder. Those that went straight down, went out at the exhaust port. Those that were not in range with the exhaust port were blown against the sides of the cylinder and picked up by the piston to do their mischief along the side walls of the cylinder. I took a coarse file and rounded the nipple over and this cured the trouble. The new scratches were not so bad and they were soon eliminated by the rubbing of the piston.

LAEN H. WILLIAMS.

Philmont, N. Y.

Brass Boiler-feed Pipes

We have five water-tube boilers set with flush fronts. The products of combustion, after leaving the tubes, pass through an opening in the side of the smoke box into a flue extending back along the side of the boilers to the uptake. With this style of return, smoke flue and setting, the boiler-feed pipe has to pass through the front wall into the smoke flue to the boiler, which it enters through a brass bushing situated about 2 inches above the top row of tubes.

About eighteen months ago we changed our boiler-feed pipe from a good grade of extra-heavy iron pipe to a good grade of brass pipe, so as to conform with one of the State laws.

This arrangement was all right and improved the appearance of the boilers, but the other day I noticed one of the pipes leaking at the short nipple close to the boiler.

After the boiler was cut out and cooled off, I started to take the pipe apart at the union, in order to put in a new nipple. I was surprised when, pulling slightly on the pipe, my wrench sank into it. On examining the pipe after it had been removed, it looked as though all the bottom of it had been worn away from the outside.

The top was of the original thickness, 3/32 inch, but the bottom was as thin as tissue paper.

In repairing this pipe we covered the part inclosed in the smoke flue with an iron sleeve and over that plastered fire clay.

I wish some of the readers of POWER would tell me what effect smoke has on a brass pipe. We burn buckwheat and soft coal mixed in the ratio of four to one and use forced draft of the steam-jet type. I would also like to know if electricity has any effect on the pipe. An electric road runs close by the boiler house. Besides this, our own ground detector is connected to the water pipe instead of being carried to the ground.

GILBERT GEICRIN.

Adams, Mass.

Drilling a Plate

I had a job of drilling twenty-eight 5/16-inch holes in 3/16-inch steel, 12 feet long by 3 feet wide; I could not get it under the drill press nor use a ratchet drill so I used a carpenter's brace. It was very hard work and it took five minutes for each hole. A new idea struck me. I center punched a hole, then ran the drill down to bottom of center-punch mark, then punched it again and ran the drill down again and so on until through. It took just one minute to go through a hole in this way and it was about as easy as drilling into wood.

HENRY N. WINANS.

Brooklyn, N. Y.

Vertical Engine Bearings

Some two years ago I had an experience that caused me to decide that in all cases where I have any authority, all vertical engines must have the shaft bearings adjusted to a good working fit; let the lead, compression, speed or any other factor be what it may, the shaft bearings must be right. I was running, at that time, vertical blowing engines. They were in the worst condition of any engines I have ever seen running, and I am very glad that six weeks was the length of my stay.

These engines were built with the shaft running in the bottom of the frame; the steam cylinder was next above the shaft and the air cylinder above all. The two cylinders were far enough apart to allow the crosshead to be fitted on the piston rod and to have room for the full stroke. The crosshead was long enough to reach from outside face to outside face of the flywheels, of which there were two, one on each side of the engine.

The engines were run at a speed of from 26 to 32 revolutions per minute and as long as the air pressure was normal nothing unusual happened. Upon two occasions the furnace got cold and the boss ordered the pressure to be run up to and held at 19 pounds per square inch. At this pressure, when the piston was on the down stroke and almost at the end, the pressure in the air cylinder was sufficient to overcome the power being developed in the steam cylinder and the kinetic energy in the flywheels completed the stroke. This caused the flywheels to ride up, so to speak, on the crank pins. As soon as the dead center was passed far enough so that the crank pins were out of line with the center of the flywheel, down came the flywheels and

shaft with a bump into the bottom of the bearings. The shaft had over an inch of play, and I can tell you that 26 of these bumps per minute were more than sufficient to show me that vertical-engine shaft bearings should be kept "swung" just as carefully as those of horizontal engines.

HARRY W. BENTON.

Cleveland, Ohio.

Water-column Guard

In almost every issue of POWER we read of some new safety device for boilers. There are high- and low-water alarms, safety valves and all the rest. I have never seen anything published, however, about devices to guard firemen against being injured by the breaking of a water glass. Oftentimes we hear of a case in which a fireman has almost lost his eyesight or in some other way been more or less seriously injured.

A device which could be easily made and one which would afford the fireman a large degree of safety is here described:

A piece of light-gage sheet iron or copper is cut to proper length and bent to a radius slightly larger than that of the glass. A slot, ¼ inch or more wide, is cut down the center so that the water level can be easily observed. With an arrangement like this fastened in front of the water glass the fireman has the benefit of considerable protection at a very small cost.

I do not claim that my idea is new, but I have never seen it described before. It is entirely probable that others can suggest devices far superior to this one and I hope they will bring them out. I do not think it is possible to have too much safety about the boiler house.

HENRY HERBERT CARTER.

Mohawk, Mich.

Adjustable Bearings

In the accompanying drawings is illustrated a device which I think will take the place of the so-called solid bearings. We have here two small engines of the vertical type which are equipped with solid bearings. These bearings give continual trouble, for within a week after new ones are put in, the wear begins to tell with a nerve-grating knock. Then one of the balance wheels must come off and the new bearing must be driven in and scraped for a half day, the result being a bad job at best at the end of the first day's run.

Perhaps it might be possible to use a bearing with a tapering thread, the general appearance of which is represented at *F* in Fig. 1. At *E E* a slot is cut the full length of the bearing. Detail *A* shows the top of the bearing in which *E* is again evident

To the right of the top view is shown a method of locking the bearing after it is adjusted; *B* is a set screw, *C* a plate which the set screw holds fast to the frame of the engine and *D* a slot in the plate.

In Fig. 2 the bearing is shown screwed in place, *K K* being clearance spaces. Fig. 3 represents another way of doing the same thing. In this drawing *W* is the top of a piece of metal, the cross-section of which is shown at *X X*.

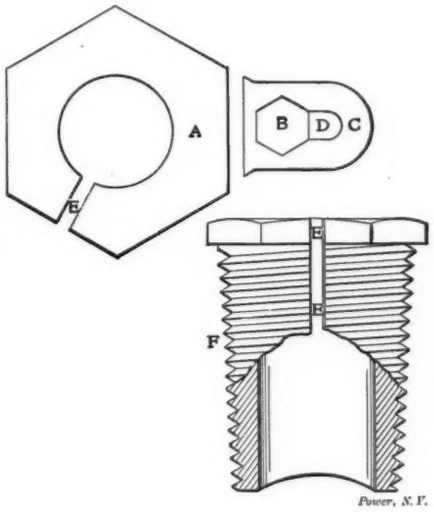


FIG. 1

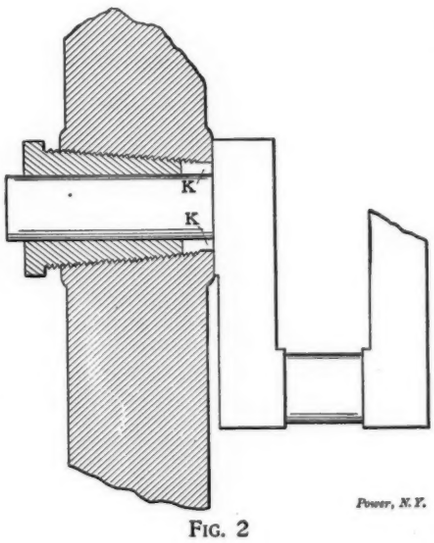


FIG. 2

Now, if the crank case be threaded to receive this piece, a tapering hole cut to receive a slotted tapering bearing, the cross-section of which is represented at *Y Y*, it is clear that any wear may be taken up by applying a wrench to the hexagon part of *X X* in such a manner as to force the tapering bearing farther into the crank case. This adjustment may be made while the engine is running.

If the cranks are running in a bath of oil, the slot *E* will help to keep the bearing oiled, and in the case of the taper-threaded bearing, the leaking of the oil may be stopped with a piece of lead driven in the slot next to the balance wheel. The lead being soft, it will be compressed by the closing of the slot when the bearing is screwed farther into the crank case.

EARL MANLOVE.

South Charleston, W. Va.

Opening a Clogged Water Column

One day the pipe leading from the boiler to the water column on one of our boilers became clogged so that it was impossible to determine the water level. It so happened that we were able to cut out this boiler without interrupting our service though, of course, it would have had to be cut out anyhow.

The boiler was emptied and every effort made to get at and remove the obstruction without success. The clogged pipe entered the boiler down among the flues where we could not get at it, and the piping was buried deep in the brickwork of the setting so that it was impossible to get at the plugs in the connection if there were any there. The piping of the column was so arranged that the whole thing would have to be taken down in order to get at the end which was clogged and if it were taken down the obstruction was just as liable to be found in the pipe at right angles to the boiler and buried in the brickwork. After considering everything, it was decided to try putting a valve in the steam connection so that it could be closed and pressure could be applied to the obstruction in the water connection. A valve was, therefore, put in the steam connection and the lower manhead put in and the boiler filled to the proper distance above the tubes; then the other manhead was put in and steam raised. When the pressure reached about 70 pounds the obstruction was readily blown out by closing the valve that had been put in the steam connection and opening the blowoff to the water column.

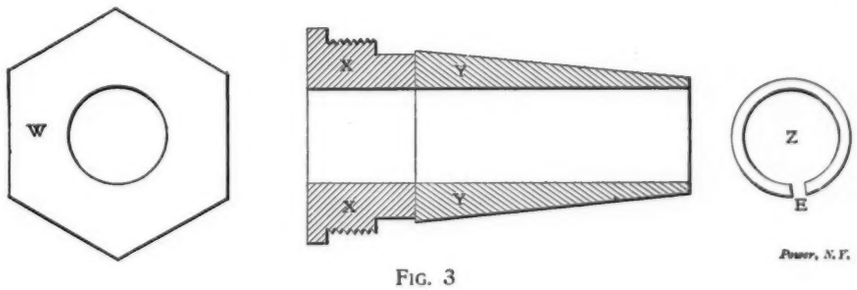


FIG. 3

This experience impressed upon me the importance of having the water connections to the water column as accessible as possible and also the importance of blowing out the water columns frequently and thoroughly. Had our engineer done his duty in the line of blowing out the water column, I am confident this trouble would have never occurred.

G. E. MILES.

Denver, Colo.

Diagrams for Criticism

In Fig. 1 are shown diagrams from a 22 x48-inch Brown engine running at 60 revolutions per minute. The engine had been in service 10 hours a day for 21 years at the time the diagrams were taken.

After some changes had been made in

FIG. 1. BEFORE VALVES WERE SET

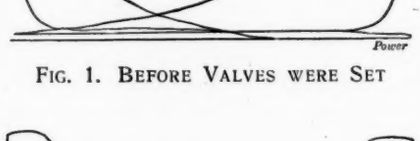


FIG. 2. AFTER VALVES WERE SET

the valve gear, the diagrams shown in Fig. 2 were obtained.

I would be glad to have readers discuss these diagrams and give their opinions of any improvement shown.

R. L. BLACKMORE.

Columbiaville, Mich.

Air Compressor Capacity

In a certain plant, where the demand for compressed air had grown so rapidly that the supply was merely enough to meet requirements, a motor-driven, cross-compound, 16 and 10 by 14-inch air compressor, delivering air at 100 pounds pressure, was down and out on account of a broken back head on the high-pressure cylinder. As air was a necessity, something had to be done quickly. Patching the head was practically impossible as it was cored out for a water jacket and also contained the mechanically operated intake valve and the check-discharge valves.

Obtaining the delivery of a new head from the makers was a problem which finally solved itself in about two weeks. There was an abundant supply of sheet-steel scrap around the yard, from which a blank flange was quickly cut out, drilled for free passage of air into and out of the head end of the cylinder, and for inlet and outlet for jacket water, and bolted on in place of the broken head.

The machine was started up and worked very well; in fact, the results surpassed the expectations of the mechanic who proposed the repair. The low-pressure cylinder was taking almost the same volume it would take under ordinary circumstances, and compressing it to a higher pressure before discharging it to the receiver, on account of the high-pressure cylinder taking air from the receiver only on the back stroke. The relief valve on the receiver was set a little higher, and although the machine was out of balance, it was hardly perceptible in the running and the machine ran for the two weeks without a hitch.

The question is, at how much less than its full rating did the machine operate? It took a little less air into the low-pressure cylinder on account of the increased clearance space. The receiver, being filled with air at a higher pressure, which expands on the intake stroke, would require less new air to fill the piston displacement. This might prove an interesting problem if worked out and explained by some of our better-educated brothers.

WILLIAM PRICE.

Philadelphia, Penn.

Mechanical Genius

At the first place in which I was employed there was an eight horsepower gasolene engine which was used to pump water for the office buildings. The engine was run one day each week and carried probably three- or four horsepower load. The man before me had taken it apart at one time or another, and did not get the gears back right, by a few cogs. He burned 800 gallons of oil in seven months. On adjusting things I reduced the oil to 14 gallons a month.

I noticed a place where he had soaked a cloth string in paint and wrapped it around a valve in the heating system. Thinking the valve was broken, I started to replace it and found it leaking around the stem. This was his way of packing a valve.

At the next place, they had a 50-horsepower boiler and a 50-horsepower engine to run a laundry, using probably five or six horsepower. The boiler had a 3½-inch lever-type safety valve that leaked. To get the steam out of the room, a heavy bushing was screwed in and a 1-inch pipe run from it out through the wall.

The stuffing boxes on the engine are of the sliding-sleeve design with two draw bolts. The valve-stem packing gave out, and the genius who reigned before my advent, had wrapped flax rope around the stem outside of the sleeve and then had cut an oval piece of tin, with holes in it, for the bolts and stem. Slipping the tin around the stem and over the bolts, he drew it up against the packing.

He did not believe in lead or cushion for he had the eccentric set back far enough to have negative lead. After changing the eccentric I was surprised to find the valve O. K. for the engine has been 30 or 40 years in service. The boiler has been used for the same length of time. With a light load, and practically no safety valve, isn't it queer a "mysterious" explosion did not occur?

FRED ECKLEY.

Hesperus, Colo.

Trouble with a Heating System

We were heating by the direct system with exhaust steam supplemented when necessary by live steam, the exhaust from some pumps and an automatic engine being sufficient except in very severe weather. The exhaust line was bypassed around an open, feed-water heater, but in order to get the benefit of an oil separator attached to the heater, we closed the bypass and passed the entire exhaust through the heater on its way to the distributing mains.

One moderate day early in spring while I was seriously considering a reduction in the accustomed back pressure of eight pounds, the manager came rushing in and demanded to know why the steam had been shut off. The relief valve was closed and the pressure gage back against the stop pin. It is true that we were running very light, having but three pumps in operation, yet the exhaust from these I judged to be more than sufficient in such moderate weather. I hastened to try the stop valves in the basement of the main building as meddlesome fingers had been at work there before. One time someone connected the high-pressure water system into the city mains to the consternation of the meters. The valves were all right but perfectly cold.

I turned on the live steam and brought the pressure back to eight pounds.

An investigation beginning at the heater was instituted but it did not last long. I accidentally touched the lever of the float operating the feed valve. It raised and closed the valve instantly. The valve had been wide open and an inch and a quarter stream of cold water under 40 pounds pressure had been going over the heater pans.

It is probable that the valve stuck until the water in the heater was lowered considerably and then opened to its full width and stuck again.

I removed the packing from all the automatic valves and repacked them, resolving to see that the feed-water heater did not again act as a condenser when exhaust steam was a valuable asset.

LOE ADDY.

Newcomerstown, O.

SUBJECTS *under* DISCUSSION

COMMENT, CRITICISM, *and* DEBATE *upon various* ARTICLES, LETTERS *and* EDITORIALS *which have* APPEARED *in* PREVIOUS ISSUES

Cement Testing Needles

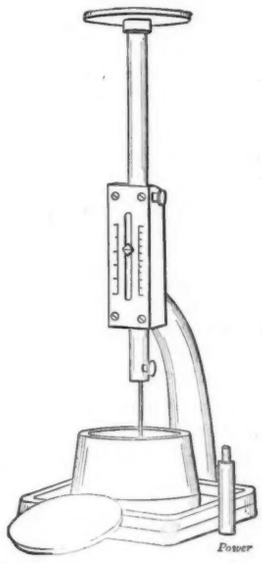
The standard method of determining the time of set of a cement, as recommended by a committee of the American Society of Civil Engineers in 1904, and the one which is almost universally used in the United States, is briefly as follows:

About 500 grams (1.1023 pounds) of cement are placed on a mixing slab, preferably a piece of plate glass. A crater is formed in the cement into which is poured the percentage of water at 70 degrees Fahrenheit necessary to produce the standard plasticity. About 20 per cent., by weight, is the amount usually required. When the water has been absorbed the paste is thoroughly kneaded with the hands (protected by rubber gloves) for 1½ minutes. The paste is then formed into a ball, forced into the large end of the hard-rubber ring of the testing machine, smoothed on the bottom, placed on a 4x4x⅛-inch piece of plate glass and the excess material cut off the top and smoothed with a trowel without pressing or ramming. A trial to see whether set has occurred is made by placing the glass and ring containing the specimen under the Vicat needle, and lowering the needle until it just touches the paste and then quickly releasing it. Initial set occurs when the needle penetrates to within 5 millimeters (0.2 inch) of the bottom; hard set occurs when the needle does not sink visibly into the paste. The time of setting is the time which elapses after the addition of the water until these events occur.

The Vicat needle mentioned above, one form of which is here illustrated, consists of a steel needle, 1 millimeter (0.039 inch) in diameter, fixed in the end of a rod, the whole weighing 300 grams (10.58 ounces) and supported in a vertical position by a stand. The hard-rubber ring is a frustum of a cone, 7 centimeters (2.76 inches) in diameter at the large end, 6 centimeters (2.36 inches) in diameter at the small end and 4 centimeters (1.57 inches) high, resting on a glass plate ⅛ inch thick. The amount of penetration is read from the graduated scale.

A plunger 1 centimeter (0.39 inch) in diameter replaces the needle and a lighter disk is used, to keep the total weight at 300 grams, when the instrument is used to determine the standard plasticity or "normal consistency" of a cement. This is the percentage of water it is necessary to use to cause the plunger to penetrate the paste, when mixed by the method described for the set test, a distance of 10 millimeters (0.39 inch) from the top surface. In cement tests all neat cement pastes are mixed to normal consistency.

Specifications generally require that the time of initial set shall be more than 30 minutes and the time of hard set more than 1 hour and less than 10 hours. This is to allow sufficient time for the placing of the concrete in the forms or the use of mortar on the work so that initial set does not occur before the material is finally placed, as any working of concrete or mortar after set has started reduces the strength. A very slow setting cement may impede the progress of the work by requiring the forms to be kept in place an excessive length of time.



VICAT CEMENT-TESTING NEEDLE

The method of determining the time of set with the Gillmore needles, as described in January 18 issue by A. S. Merrill, has been superseded in general practice by the above method as being more accurate; but the Gillmore needles have been retained in the United States army specifications. The needle, to the use of which Mr. Merrill takes exception, weighing 2½ pounds and having a flat end 1/16 inch square, is recommended by the Engineering Standards Committee of Great Britain in the "British Specifications for Portland Cement."

G. WISE.

Philadelphia, Penn.

What Causes the Fittings to Break?

I think if S. D. Seeley will put in an expansion swing joint above and below the tee that broke, made up of ells and nipples at right angles with the run, he will cure his trouble. The break was undoubtedly caused by expansion and contraction. In his diagram the ells are shown parallel to the run, which makes the construction too rigid. The manhole with the union in it is best suited for the change, or if the manholes are far apart make the change in both.

What does a plumber know about the effects of steam? It's out of his line entirely.

W. H. H. PLOWMAN.

Philadelphia, Penn.

Misleading Statements in Advertising Matter

In the issue of February 1, William Westerfield takes exception to a statement which he saw in a recent advertisement, to wit: "Exhaust steam will give up its heat faster to the radiating surfaces than will live steam from a reducing valve. Live steam that has passed through a reducing valve is in a superheated condition; that is, it is a gas, and gases are poor conductors of heat. That is one reason why you superheat steam for the engine, so that it will not give up its heat so readily to the cylinder walls."

Mr. Westerfield is, in my opinion, misinformed when he replies as follows:

"The best authorities agree that the rate of heat transfer depends upon the relative temperatures of the heating agent and the agent to be heated, no matter what they may be." In this he is wrong for the authorities do not agree that such is the case.

If Mr. Westerfield will heat a piece of wood and a piece of iron to the same temperature, say about 150 degrees Fahrenheit, and then touch each in turn, he will find that the iron feels hotter than the wood because, although its temperature is the same, it gives up its heat more rapidly than does the wood, for the reason that it is a better conductor of heat. If this test is not convincing, let him thrust his hand into a steam jet fed with dry steam at 175 pounds pressure. The steam, when striking his hand, will be at atmospheric pressure and at a temperature of about 375 degrees, which gives the steam a superheat of 163 degrees. Then let him put his hand over

the spout of a tea kettle in which water is boiling briskly and see which feels the hotter. The superheated steam, while being 163 degrees hotter than the water vapor from the kettle, strikes the hand without burning it, while the vapor from the tea kettle will scald the flesh for the reason that, while the temperature and specific heat of the latter are lower than those of the superheated steam, it is a better conductor of heat.

Mr. Westerfield says, further: "We do not superheat steam so that it will give up its heat less readily to the cylinder walls, but we superheat it in order to avoid some of the loss of power due to initial condensation. It would be reversing well accepted thermal laws to say that with greater temperature range the transfer of heat is less than when the range is low. If we superheat steam sufficiently to enable it to part with some heat without condensing on entering the cylinder, we avoid a considerable loss, etc." These statements are contradictory, insufficient and incorrect. Mr. Westerfield's conception of the relative action of saturated and superheated steam in an engine cylinder is evidently somewhat vague. He apparently forgets that the steam engine is primarily and exclusively a heat engine operating on a given cycle, depending on its construction. We do superheat steam so that it will give up its heat less readily to the cylinder walls, in consequence of which we avoid the losses due to condensation before cutoff. The object of superheating is to secure dry steam in the cylinder, and the actual gain which follows the use of superheated steam is due to the more or less complete removal of loss by cylinder condensation. When the working fluid is saturated steam, no transfer of heat can take place from the steam to the cylinder walls without an accompanying deposition of water which during the exhaust stroke is evaporated at the expense of the heat given back by the cylinder walls. The result is that the mean temperature of the cylinder walls with saturated steam is much below that of the steam on entering the cylinder. On the other hand, when the steam is sufficiently superheated it is in a far more stable condition than before the superheat was added. It approaches more nearly the condition of a perfect gas, in which state it is a poor conductor of heat. In this state it can part with all of its superheat without undergoing liquefaction. During liquefaction water and steam are at their best as conductors of heat.

The drier the steam is at cutoff, the more work is done per pound of working fluid passing through the cylinder. The drier the steam is at release, the less is the demand upon the cylinder walls during the exhaust stroke for heat for re-evaporation, and the higher is the mean temperature of the cylinder walls.

Superheating thus removes the principal source of loss of heat by the walls, namely, water in the cylinder at release; and it reduces also the amount of the heat exchange between the steam and the cylinder.

F. H. COBLE.

Baltimore, Md.

Beaded Boiler Tubes

The boiler explosion at Reading, Penn., described in the January 25 issue, prompts me to say a few words. I hope that all readers of POWER will turn back to the description, read it carefully and in the future, when having new tubes put in, they will have them annealed and beaded down in good shape. I have been at that work for 25 years and have had many hot disputes over this question. If I am not permitted to bead them down, I will not put them in. Many contend they are just as strong and hold just as well without beading, but this is not so. I do not wish it to be understood that I claim beading to be an absolute preventive of explosions; but I do claim that the work looks better, lasts longer and is stronger. Some boiler makers say it is only an extra expense, but I think it makes a rather high first cost, if they have to pay damages as a result of slighting their work.

In conclusion, let it be remembered that a thing worth doing is worth doing well.

C. T. PERRY.

Oil City, Penn.

Homemade Ejector

I was much interested by the description of the homemade ejector described by Mr. Regan in the January 25 number.

I would be glad to have Mr. Regan or anyone else give some suitable dimensions. Is it necessary to make dimensions *A* and *B* any special length, or may they be made to suit conditions?

Would the ejector work on a small steam pressure, say, 8 or 10 pounds; if so, how big a lift would it handle?

If this ejector will not work on 8 or 10 pounds of steam, can any reader suggest one that will? One which would be equally as cheap.

C. A. WILLARD.

Lynn, Mass.

While I do not doubt for a moment that a homemade ejector would work if constructed as per sketch printed in the January 25 issue, since Mr. Regan saw it in a power plant, yet I confess that I cannot see what keeps the steam from rushing out through the line of least resistance, the lower pipe, and out at *D* and thus bypassing the nozzle without creating any vacuum at *B* or at the suction pipe. Will you kindly tell us, Mr. Regan?

I made several ejectors with tees and ¾-inch angle valves. They work the best when steam is supplied at 50 pounds or water at 45 pounds per square inch. I made the inlet of ½-inch pipe and the suction and discharge of ¾-inch pipe.

ALEX. DOLPHIN.

Jamaica, N. Y.

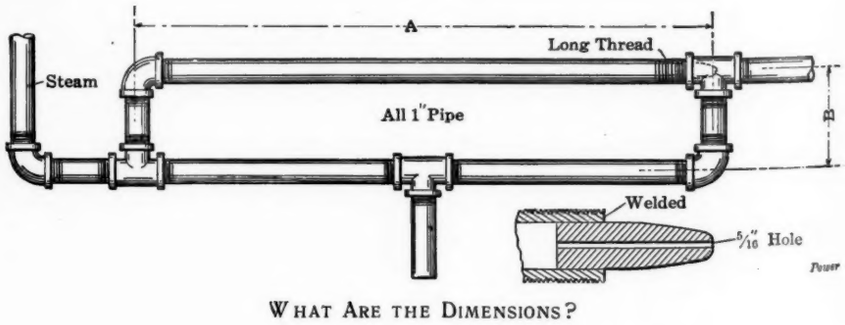
Boiler Safety Appliances

In regard to the editorial on "Boiler Safety Appliances" in the January 11 issue, I beg to differ with the writer on some of the items.

The placing of valves between water column and boiler can hardly be silly or it would not be allowed at all by the State of Massachusetts nor by the steamboat inspectors, nor the British Board of Trade.

The Massachusetts rules call for valves in this position being sealed open so they cannot be meddled with, but does not prohibit them, and they can be used in case of emergency. Not only in case of a broken gage glass, which is a minor difficulty, but in case of a stoppage of valves or connections, when the position of the obstruction can be found without shutting down the boiler.

It must be remembered that a water glass on a boiler is merely a convenience, not a necessity, and for absolute reliability the placing of cocks directly on the shell is necessary and should be made compulsory.



WHAT ARE THE DIMENSIONS?

Drain pipes from the bottom of the column proper should not be less than 1 inch, as I have found that when smaller pipes are used the column is not cleared when the boiler is being blown down.

J. Y. GILROY.

Westchester, N. Y.

No Heat without Circulation

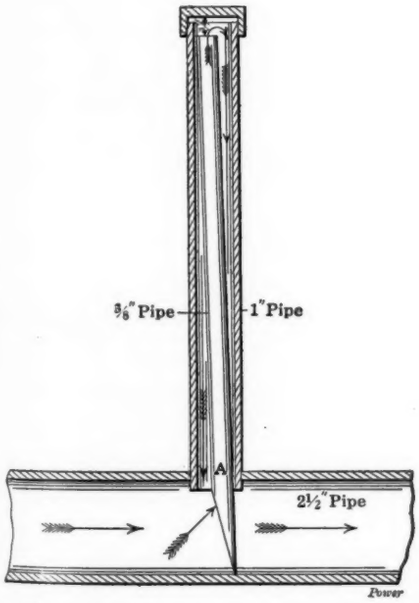
In the January 25 issue I notice W. Chinsano's inquiry relative to the heater which does not heat as he expected it would.

The heater fails to heat because of insufficient circulation, the upper part of the vertical pipes being full of air, which prevents the steam from going to the top.

As a remedy for the difficulty I advise that he procure some ⅜-inch pipe and put pieces inside of the 1-inch pipe as shown at *A*.

Cut the ⅜-inch pipe at an angle and let it rest on the bottom of the supply pipe and extend to within half an inch of the top of the vertical pipe.

The result will be that the steam will enter the internal pipe at the bottom, as indicated by the arrows, and heat it to a



METHOD OF PRODUCING CIRCULATION

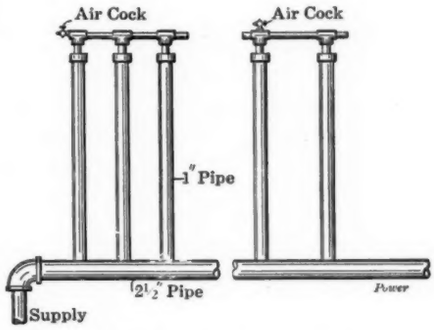
higher temperature than the 1-inch outside pipe.

The force of the steam in the inside pipe, due to the difference in temperature, will displace the cooler and heavier air and vapor surrounding it and thereby maintain an even circulation.

J. W. PARKER.

Clinton, Mass.

Mr. Chinsano has made no provision to relieve the radiating tubes of the air which they contained before the steam was turned on. The air being heated rises to the upper ends of the tubes. I suggest that he tap a hole in one of the caps and screw in a small pet-cock. This will release the air and the tube should heat throughout its full length. After seeing the effect on one tube, I suggest that he tap the caps of all the tubes. It would be quite expensive to put separate valves on each pipe, so the best way is to connect a number of caps



RELEASING AIR FROM VERTICAL PIPES

with a line of tees and nipples and put an air cock at the end of the line. One-eighth-inch fittings will be plenty large enough.

It may be necessary, on such a large radiator, to use more than one cock, so one could be placed in the middle of the radiator and in place of a tee, a cross could be used at the point where the extra cock was inserted.

I trust that if any reader sends in a suggestion which is used by Mr. Chinsano he will let us know of his results. These practical letters are of great interest to us all and we oftentimes wonder how an engineer comes out of his trouble, and which suggestion, of the many printed, is followed.

JAMES ANDERSON.

East Boston, Mass.

I think I can suggest a way for Mr. Chinsano to overcome the trouble with his radiator. He has shown the pipes arranged in such a manner that there is no way for the air to get out of the pipes.

In order to get steam into the 1-inch pipes, he must first get the air out of them. He can do this by putting into each pipe a radiator, air-escape valve, or what will amount to the same thing, he can connect all the top ends of the upright pipes by a horizontal pipe and use one or more air valves.

I would suggest he connect only pipes of the same hights together or at least not try to make the air run down hill.

He can try the effect of letting the air out by drilling a hole in the top end of one of the pipes and turning on the steam. If this proves effective in getting this one pipe to heat up, then the same thing will work for all the pipes and they can be fixed as stated above. No radiator will heat up unless the air can get out of it.

P. S. WITHENBURY.

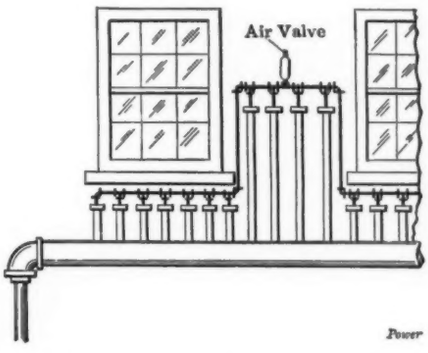
East Lansing, Mich.

If Mr. Chinsano will insert sheet-iron strips in the 1-inch pipes and let them extend to within 2 inches of the caps and down into the 2½-inch pipe and place an air vent on the last pipe; that is, the one at the end opposite to the supply he will get over his difficulty of lack of circulation. This is the same construction as that of the Nason radiator which has no return bends.

CHAS. SEAMAN.

Memphis, Tenn.

To my mind, the above heading which has been given to W. Chinsano's inquiry, in the January 25 number, regarding the failure of his radiator to heat up uniformly, owing to the method of construction, is misleading. While the trouble which he is experiencing can truthfully be said to be due to lack of circulation, the cause of this lack of circulation is the fundamental defect, and if there was nothing but steam allowed to get into his manifold it would probably work perfectly, providing, of course, that the proper pipe areas are used for supplying and draining the manifold. Under such conditions the condensation of the steam in the vertical pipes would produce practically perfect circulation. Mr. Chinsano's difficulty is caused by one of the simplest and probably the most frequent of



AIR VALVE PIPED TO RADIATOR LEGS

troubles to which heating systems are subject, and this is trapped air. Every radiator on a heating system generally requires an air valve to overcome just the trouble that is being experienced with Mr. Chinsano's layout, and with a manifold built as he shows it, each of the 1-inch vertical pipes is in effect a separate radiator and would require an air valve on its upper end to make it work perfectly. This, of course, would not be a practical remedy from the fact that the expense would be beyond all reason. The cheapest and most effective method

that could be adopted, would be to connect the top ends of all the vertical pipes together and let a single air valve serve for an entire section between two windows, as illustrated in the figure. The connecting pipe may be ¼ or ⅛ inch in size as it will not be required to carry steam but simply permit the escape of the air in the system when starting up, and keep the pipes free from air during operation.

J. E. TERMAN.

New Haven, Conn.

Condenser Question

In the issue of December 21, 1909, J. S. Labonne asks a question on condenser operation.

He appears to be operating a Knowles injector condenser and having found that it works satisfactorily with the tail pipe open to the atmosphere wishes to learn the reason for arranging these condenser tail pipes with the discharge end under water.

Mr. Labonne does not state what type of plant this condenser is working with, but seeing that he finds the machine works satisfactorily with tail pipe open to atmosphere it may be assumed that his engines run on a steady load.

If the load on the engines was reduced with the result that the steam consumption was also reduced, a smaller quantity of injection water would be required in the condenser to maintain the required vacuum.

On the load being further reduced, conditions would reach a point where the condenser would fail to keep a full bore of water in the tail pipe. If under these conditions, the tail-pipe discharge was open to atmosphere it is quite clear that owing to the lower pressure existing in the condenser the external air would flow up the tail pipe and so destroy the vacuum. If the tail-pipe discharge was sealed in the usual manner this, of course, would be impossible.

W. VINCENT TREEBY.

Ilford, Essex, England.

Traction-Engine Injector Troubles

In the January 25 issue of POWER, J. M. wants to know the cause of his injector breaking while his engine is under way. I have had several years' experience with traction engines and will give my experience with injector troubles.

An injector will break for several reasons. The water may be too low in the supply tank, the screen may be clogged up, or the steam pipe may not be connected to the boiler at the right place. If the steam pipe is connected to the shell of the boiler about 18 inches from the rear, when going over rough roads or climbing a hill, water will occupy the space intended for steam and cause the injector to be fed with water or wet steam instead of dry steam. The remedy for this cause of trouble is to place the injector steam pipe in the steam dome, or as near as possible to the center of the boiler. It is a good policy not to carry too high a water line, especially if pulling a heavy load, as this is apt to cause the boiler to prime. When priming takes place, the injector has a tendency to refuse to work.

A. C. KASTNER.

Winnipeg, Man.

Drop in Expansion Line

William Westerfield's experience with a Corliss valve gear, as related in his article, "Experiences of a Trouble Man," reminds me of an experience I had with the high-pressure cylinder of a tandem, compound Corliss engine.

I took a number of diagrams when the engine was new and they had the same sharp drop in the expansion line which Mr. Westerfield's diagrams show.

I tried every change of valve setting listed in the "farmers'" almanac before taking out the exhaust valves to examine them. When I did take them out I found that the valve had a bearing of only ⅛ inch and then a hollow. After dressing the valve down to a good bearing I took new diagrams and found that the drop had disappeared.

I think the hollow in the valve gave the steam a chance to blow by quite rapidly for an instant and thus the drop appeared on the diagram. This is my experience with one engine; it may be taken for what it is worth.

I like to see all these little kinks discussed as it is helpful when one gets up against trouble.

W. R. CATON

Lisbon Falls, Me.

Receiver Pressure

On page 1032, December 21 issue, E. H. Lockwood explains why the practical engineer is wrong in supposing that lowering the receiver pressure and shortening the cutoff in the high-pressure cylinder results in a saving of fuel. He states that it is just as important to have a cutoff in the low-pressure as in the high.

In 1871, Ransom came out with his siphon condenser and a manufacturer, "Tom" Wallace, had one put on his 24x 48-inch engine which resulted in the saving of about 30 per cent. of the coal.

Mr. Wallace reasoned that if a vacuum would save that amount on a 24-inch cylinder, it would save a lot more on a 30-inch one; so he put on a 30-inch cylinder. This resulted in raising his coal bill to about what it was with the 24-inch cylinder, noncondensing.

It so happened that his load with the 24-inch cylinder would produce a diagram something like that shown by the full lines, Fig. 1, and atmospheric pressure doing work up to the point *A*, while with the 30-inch cylinder the cutoff would allow it to do work up to the point *B*, so that the vacuum was not as useful with the 30-inch as with the 24-inch cylinder. If he had dropped his steam pressure, which in this case corresponds to receiver pressure, to the point *C* so as to keep up the terminal pressure the story would have had a different ending.

Mr. Lockwood brings up the temperature range in the high-pressure cylinder. We have been taught that a wide temperature range in a cylinder means a lot of condensation.

A cylinder is better protected from radiation than an ordinary steam pipe. We are taught that if steam be expanded in a steam pipe by throttling or any cause that the total heat remains and results in superheating the steam.

Why does not the same law apply in expanding steam in a steam-engine cylinder?

W. E. CRANE.

Duluth, Minn.

Writing for Catalogs

Noticing the advertisement of a device that would be of advantage in the plant where I was employed, I wrote for a catalog, and, giving the size desired, asked about the probable cost. Not wishing to have the salesman call at the plant I gave my house address and told my wife that in case a salesman called to tell him that I did not want him to call at the plant. A salesman came, and my wife delivered my message.

What did he do? Went back to his office and from there called up the plant by telephone, inquired for the chief engineer and told him that he learned that we were in need of such a device.

This so prejudiced the chief, the suggestion having come from one under him and through someone else, that the salesman never made a sale. Instead, we have gone to more expense to repair the old device than the new one would have cost. All this is due to lack of tact on the part of the salesman.

My object in inquiring the price was to be able to give the probable cost when I recommended the device to the chief and I am sure that I could have induced him to put one in. Instead the chief now looks upon me with suspicion, thinking, perhaps, that I am after his job.

I find that salesmen are a nuisance anyway. The engineer that knows how to read and write and subscribes to power-plant magazines knows what he wants before a salesman sees him. He can and does order by mail or telephone. By so doing he saves the time that a salesman, or a number of them, would

take up. I used to have a sign over my engine-room desk which read: "My time is worth money, what is yours worth?" and I found it useful to point to it sometimes to get rid of them.

I prefer to write to the "houses" for quotations and then award the orders at my leisure, unhampered, and to the man who sells the best goods at the lowest price. Any other method is unbusinesslike and an engineer must be a business man and get the most he can out of everything and everybody for the money he spends.

W. F. MEUNER.

Brooklyn, N. Y.

Don'ts for Engineers

I read the comments in the January 25 issue by J. O. Bender on some of the "Don'ts for Engineers," by H. E. Wood, published in an earlier issue. I also read the above mentioned "Dont's" and was very much pleased with them.

Of course, an old experienced engineer who has paid close attention to his business would not take nor need any such reminders. But for the young man who perhaps is working under his first or second license, I am of the opinion that these "Don'ts" will save a great deal of trouble. At any rate, I do not think they are deserving of ridicule such as Mr. Bender poked at them. For instance, in his comment on "Don't let too much smoke escape from your stack," he says, "Brother, we just have to let it escape, there's more to follow." If Mr. Bender was in charge or even shift engineer of any plant in Chicago he would find that an excuse such as his would not satisfy the smoke inspector. They would instruct him on the rebuilding of his fire box and combustion chamber. Smoke prevention does not depend entirely on the expertness of the fireman.

In a certain plant in Chicago we had so much trouble with smoke before we found the cause that the plant came near being closed down. It turned out to be neither the fault of the firemen nor the coal. We lowered both the ashpits and grates to increase the space between the boiler and the grates and had no further trouble.

Perhaps this suggestion will aid Mr. Bender in cutting down his follow-up smoke.

A "Don't" which I would add to those given by Mr. Wood is, Don't wash a gage glass with coal oil as the glass is apt to break very shortly thereafter.

If the youngster who melted his crank-pin box had read and been governed by Mr. Wood's "Don't think it necessary for a crank-pin box to fit tight sideways," some money might have been saved for his employer.

R. P. WILCOX.

Mayville, Wis.

Pound in an Engine

After reading the discussion on "Pound in an Engine" in the January 18 issue, there came to my mind an incident which happened several years ago. I was running a traction engine for my uncle during the threshing season. The engine had just been returned from the repair shop and we thought it was in first-class condition.

One day a pound was noticed in the engine. My uncle rather prided himself on being a good engineer, but this pound had him stumped. All the "engineers" in the neighborhood declared they knew where the pound was, but none really did discover the cause of the trouble. One day a fellow workman let drop a remark which helped me to locate the trouble. I made a few experiments to be certain that I had located it and then asked my uncle to stand in front of the cylinder and listen while I shut off the steam. The pound was caused by the piston head hitting the cylinder head. After adjusting the piston travel the pound disappeared.

GEORGE G. DICK.

Berea, Ky.

Cause of Tubes Pitting

P. J. Keating, in the January 11 issue, tells about plugging a small hole in a boiler tube with a tack. This illustrates how a very simple thing may serve to tide over until a more convenient time is presented for making permanent repair. I never used a tack but I have made a tapered plug and tapped out the hole for the plug. I have also screwed wood screws into small holes through the tubes of fire-tube boilers when they were near enough to the end of the tube to permit doing so.

And now I will say what I had in mind in the first place, which deals with the cause of the holes.

Five or six years ago I installed a new tubular boiler and an engine near Seattle, on the shore of a small lake. Extending back from the edge of the lake was from one to three or four rods of marshy ground grown over with grass and small brush.

In laying the suction pipe of the pump I ran it to the nearest point on the lake and just far enough into the lake to prevent the suction from picking up decayed vegetation or trash. There was no scale-forming matter in the water, but in less than a year small holes began to pit through the tubes near their ends. Then we began to use plugs, wood screws or anything that would hold until we could get a convenient time to replace the tube. Sometimes a hole would break through too far from the end; then it was necessary to plug up the tube.

I tried to get the owner to have the water analyzed to determine what was destroying the tubes, but could get nothing done. Most everything was tried; zinc was suspended in the boiler, etc., but nothing did any good. Two or three months after the tubes began to pit, the sawdust and other trash began to accumulate about the suction pipe so that I had to move it. I ran it under the mill at an angle of about 40 degrees to the line it took in the first place and extended it 4 or 5 rods beyond all marsh land into deep water.

This ended all trouble with pitting. No more trouble was had for nearly five years. Why there was such a difference in the water at places such a short distance apart I cannot say. Perhaps some other reader of POWER can.

WILLIAM YATES.

Seattle, Wash.

Adjusting Indicator Cord

C. W. Long in the January 25 number cites the case of a cross-compound engine of 60-inch stroke to which he says the method of adjusting the indicator cord, which I outlined in the November 30, 1909, issue, would not apply. The method is, however, distinctly applicable.

For indicating this engine, probably reducing wheels would be used and there would be a pin or other projection upon each crosshead to which the cord may be fastened. The head-end center position of the pin may be marked on the crosshead guides while the engine is running. If, now, a point is located 30 inches from this point toward the crank, the mid-stroke of the pin is determined. The cord may then be made of sufficient length to reach this position when the paper clips on the indicator drum are opposite the pencil point and it may then be attached to the pin. It is not necessary to put the engine in any exact position to do this.

Mr. Long says that the method is not applicable to locomotive work. I should like to know why not? If a pantagraph or pendulum rig is used it would probably be so arranged that the pin to which the cord is tied is vertically below the fixed center of the rig at mid-stroke. If so, it is a simple matter to make the cord of such length as before described.

The point that I wished to make in the original publication is that it is easier and nicer to adjust the length of the cord from mid-stroke than from dead centers. It is obviously simpler to deal with one datum point than two. To do this it is not necessary to put the engine on either center but to find the mid-position of the stroke.

JULIAN C. SMALLWOOD.

Philadelphia, Penn.

Special Furnaces for Burning Tanbark

Designs of Furnace Additional to Those Presented by David Moffat Myers in His Original Paper and Some Discussion by William Kent and F. R. Hutton

In the November 30, 1909, issue of POWER, a paper on the above subject by David Moffat Myers was presented, and in the February 1 issue a discussion of the paper by A. A. Cary. The author has recently made the following addition to his paper and discussions by William Kent and Prof. F. R. Hutton are also available.

The accompanying illustrations, reproduced from working drawings and sketches, will give some idea of the construction of furnaces for burning spent tanbark, sawdust and bagasse.

*Presented before the American Society of Mechanical Engineers.

Fig. 1 is a working drawing of the self-feeding tan-burning furnace designed by Mr. Myers and referred to in the original article. Fig. 2 shows the construction of the grate bars, which provide the horizontal draft opening tending to produce the draft action previously referred to. Fig. 3 shows the application of these grates to bagasse burning. The stokers are done away with in this case, the fuel being fed by gravity to the feed chutes with weighted flaps which are used all over the islands of Cuba and Porto Rico. This burner has not yet been applied to bagasse burning.

Figs. 4 and 5 show the types of tan furnaces found by the writer in common use throughout the country, Fig. 4 being the old Hoyt furnace which was originally designed when tanbark was so plentiful that it was necessary to burn it. The writer has found these furnaces with inside lengths as great as 24 feet on the grate surface. Fig. 5 shows a more modern type of burner designed to give a more even distribution of the fuel on the grates.

Fig. 6 shows a more uptodate furnace designed for the hand firing of a mixture of coal and tan, the coal being mixed with the tan before entering the furnace, which is supplied with shaking or shaking and dumping grates. When coal

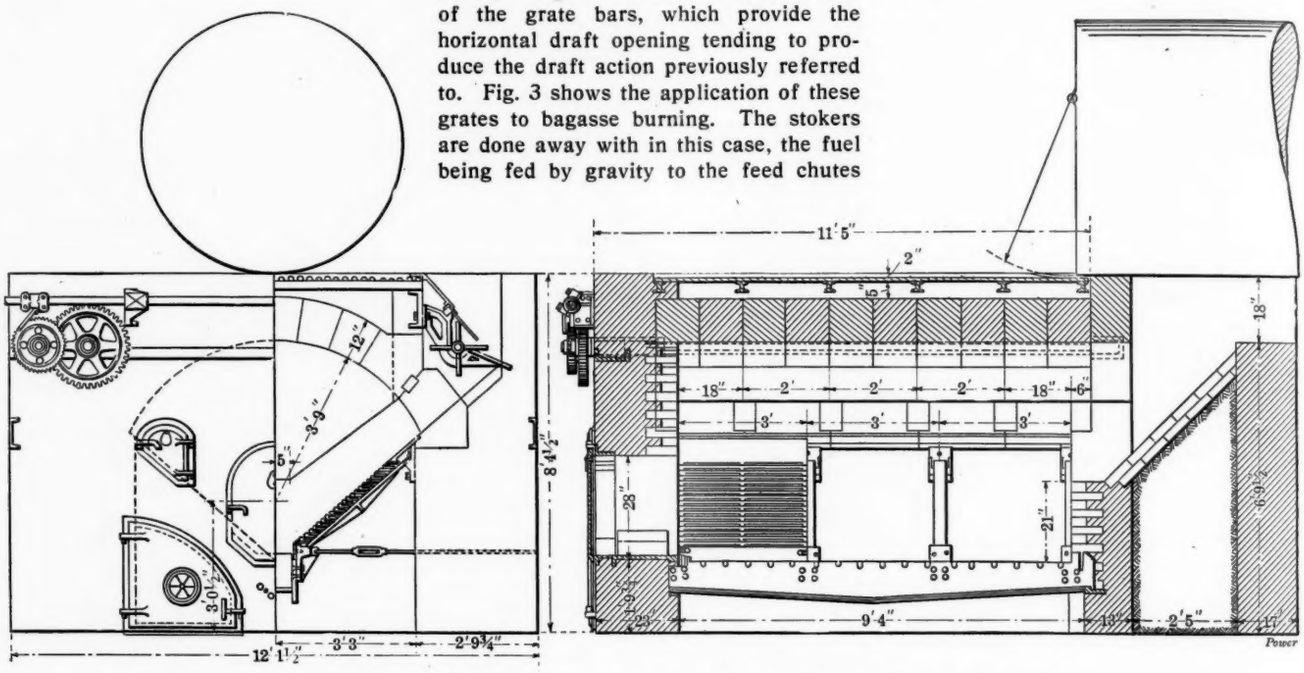


FIG. 1. END AND SIDE SECTIONAL ELEVATION OF THE MYERS TANBARK FURNACE

FIG. 2. DETAILS OF GRATE BAR OF THE MYERS FURNACE

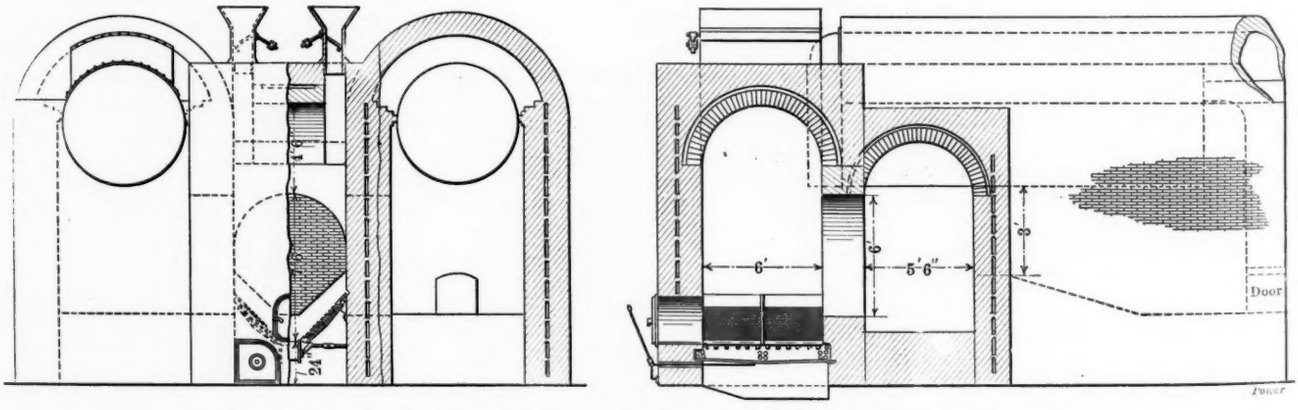


FIG. 3. SIDE AND END ELEVATIONS OF THE MYERS FURNACE FOR BURNING BAGASSE

is mixed with tan in any considerable proportion, more air is required for combustion, the best air spacing in the grate bar being found to be 3/8 inch. The percentage of draft area for this purpose should be about 40 to 50 per cent., depending upon how large a percentage of coal is used with the tan.

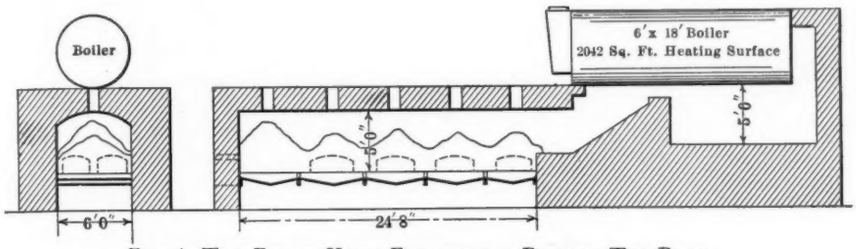


FIG. 4. THE EARLY HOYT FURNACE FOR BURNING TAN BARK.

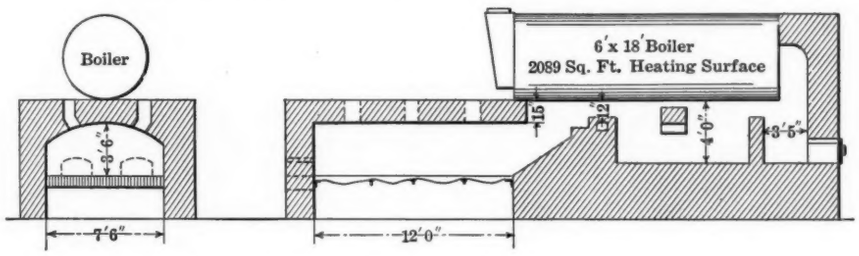


FIG. 5. A TAN FURNACE WITH SIX FEED HOLES.

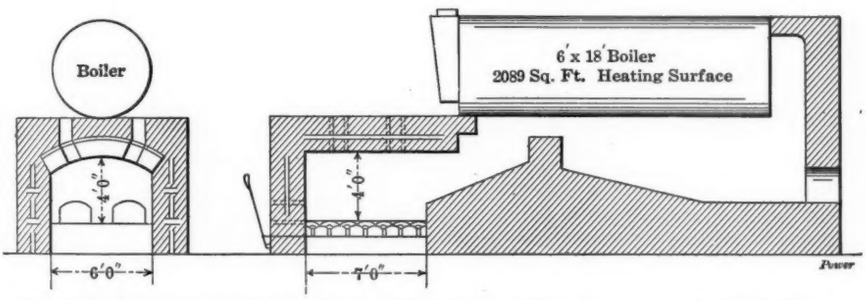


FIG. 6. A FURNACE WITH SHAKING GRATES FOR BURNING A COAL AND TAN MIXTURE.

Fig. 7 shows what is known as a hump-back grate, which has been installed in different tanneries for the purpose of increasing the consumption of fuel in a given furnace. For instance, in a plant that had trouble in consuming all its tanbark, the writer merely took out the grate bars and put in a ridge bar as shown and converted the grate surface into the hump-back form. The result was that the consumption of tanbark per furnace was increased from 12 tons per day on the dry-bark basis to 15 tons, or 25 per cent.

Fig. 8 shows what was known as the Thompson type of tan furnace. The McMurray furnace, with a convex grate surface and feed pipes, is a type of which the writer has seen quite a number in operation.

Fig. 9 is another form of tan furnace which gave good results in a plant in the South. The hump-back form of grate is reversed something like that used in the writer's stoker furnace, except that the tan is fed through a number of feed holes along the upper edges of these grates. This furnace was designed by the foreman in a Southern tannery.

Fig. 10 shows a design of the writer's for an adjustable gravity-feed furnace for burning tan or sawdust. The feed chutes are rectangular in section and contain adjustable chutes to regulate the depth of tan on the grates for any condition of draft, etc.

DISCUSSION

William Kent thought that the principal cause of poor economy in the burning of tanbark, besides the difficulty of securing good combustion in the furnace, is the amount of heat that is carried away in the shape of superheated steam in the chimney gases. If the bark, after being partly dried by compression, were further dried in a rotary drier by the waste heat from the chimney gases, there would be a very important gain in economy. He had made a calculation showing the theoretical results that may be obtained in burning tanbark of different degrees of moisture under certain assumed conditions, the results of which are given herewith:

The dry bark is assumed to have the following composition:

$$C = 0.50;\ H = 0.06;\ O = 0.40;$$
$$N \text{ and ash} = 0.04.$$

Substituting in Dulong's formula,

$$14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right)$$

its heating value is 7920 B.t.u. per pound. Bark containing 20 per cent. moisture would have a heating value of

$$0.80 \times 7920 = 6336\ B.t.u.$$

Assuming the chimney gases to escape at 600 degrees, the heat required to evaporate the water from 62 degrees and to superheat the steam to 600 would be

$$(212 - 62) + 970 + 0.48\ (600 - 212) = 1306,$$

or for 20 per cent. moisture, 261 B.t.u. per pound of tan.

The 0.06 pound of H in a pound of dry tan will unite with

$$0.06 \times 8 = 0.48$$

O, making 0.54 pound of H_2O, which escapes as superheated steam carrying away

$$0.54 \times 1306 = 705\ B.t.u.$$

for each pound of dry tan, or

$$0.80 \times 705 = 564\ B.t.u.$$

for tan with 20 per cent. moisture.

Assuming 25 pounds of air to be required per pound of C + H in the fuel, or

$$25 \times 0.56 = 14\ pounds$$

of dry tan; the heat carried away by this air heated to 600 degrees is

$$0.24 \times 14 \times (600 - 62) = 1808\ B.t.u.$$

per pound of dry tan, or 1446 B.t.u. for tan with 20 per cent. moisture. Using the figures thus found the following table is constructed:

Suppose that tan with 60 per cent. moisture were dried to 20 per cent. before being put into the furnace; using for this purpose the waste heat of the chimney gases, we would then have 0.40 dry tan + 0.60 moisture dried to 0.40 dry tan + 0.10 moisture, 0.50 water be-

Moisture.	B.t.u. per Lb. Wet Tan.	Losses of Heat Due to			Sum of Losses.	Net Heat Value, B.t.u.	Efficiency, Per Cent.	Lb. Evap. per Lb. Wet Tan.
		Moisture.	H in Fuel.	Heating Air.				
0.20	6336	261	564	1446	2271	4065	64.2	4.19
0.30	5544	392	493	1266	2151	3393	61.2	3.50
0.40	4752	522	423	1085	2030	2772	57.3	2.81
0.50	3960	653	352	904	1909	2051	51.8	2.11
0.60	3168	784	282	723	1789	1379	43.5	1.42
0.70	2376	914	211	542	1667	709	29.8	0.73
0.80	1584	1045	141	362	1548	36	2.5	0.03

ing removed. Suppose the moisture and the waste gases left the drying chamber at 300 degrees. Each pound of water dried out would take

$$(212 - 62) + 970 + 0.48\ (300 - 212) = 1162\ B.t.u.,$$

and 0.5 pound would take 581 B.t.u. The H in the 0.40 pound of dry tan would make 0.216 H_2O, which would take away

$$0.216 \times 1162 = 251\ B.t.u.$$

Heating the air would take

$$0.40 \times 14 \times 0.24 \times (300 - 62) = 320\ B.t.u.$$

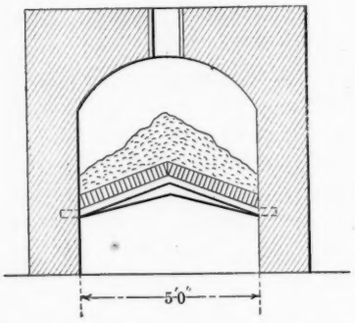


FIG. 7. CROSS-SECTION OF FURNACE WITH HUMP-BACK GRATE AND BEARING BAR

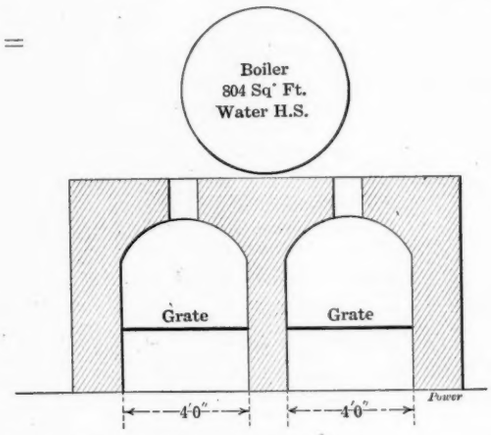


FIG. 8. DOUBLE-ARCH FURNACE OF THE THOMPSON TYPE

The sum of these is 1152, which subtracted from 3168, the total heating value of tan with 60 per cent. moisture, leaves a net value of 2016 instead of 1379, the figure given in the table. The efficiency would be

$$2016 \div 3168 = 63.6\ per\ cent.,$$

instead of 43.5 per cent. and the evaporation from and at 212 degrees

$$2016 \div 970 = 2.08\ pounds$$

instead of 1.42 pounds.

It was the opinion of Prof. F. R. Hutton that the three cardinal principles for the complete and smokeless combustion of a reluctant fuel involve the following:

Time enough for access of oxygen in the air to the carbon gas from the fuel.

Temperature enough for the rapid and complete chemical union of this oxygen with carbon and hydrogen.

Room enough for each atom of fuel gas to meet the oxygen atoms with which it is to unite. The practical attainment of these results is made more difficult when the fuel is wet and in small particles of light weight.

We have the conflicting conditions of a hot fire and a slow rate of combustion, to combine with an intensity of draft which shall not be high. Mr. Myers does this by using the step-grate idea, so as to admit the necessary air horizontally between the overlapping bars, whereby the dropping of fine fuel into the ashpit is prevented; but in addition and as a special excellence of the design, the grate is made to consist of two sections facing each other with their planes parallel to the long axis of the dutch oven and the shell of the boiler. They are, as it were, upon the inclines of a truncated capital letter V.

The bark is fed by a measuring stoker cylinder, which drops a determined volume upon the whole length of the upper bar at each partial revolution, and this fall of new material displaces downward some of what has been drying and growing ready to ignite from the previous charges. At the bottom of the truncated V is a dumping grate from which the residue of ash may be released at intervals.

The consequence of the inclination of the two grate sections, with a horizontal inflow of the air, seems to be the same as is produced in a successful form of burner for acetylene gas. The two currents of gas and draft appear to meet in the center or in the axis of the V and an intense combustion takes place there, the heat of which reverberates downward from the arch of the oven, raises the temperature of the upper layers of fuel, and stimulates the rate of the union of combustible with oxygen. Such a furnace, of course, is not subject to the alternations of the "famine and feast" conditions when excess of wet fuel deadens the fire and causes a smoky and slow combustion, alternating with high heat and good flame and followed in turn with burned-out spots in the fire until fresh charges came in through the holes.

The system is also most effective for "bagasse," the wet juicy fiber of the sugar cane after passing through the pressing rolls. This is more difficult to stoke mechanically than the comminuted bark, but the requirements for its successful combustion are very satisfactorily met. Sawdust and scrap from woodworking shops also are burned in furnaces of this design with less danger from sparks at the stack.

It should be plain that the grate surface must be reduced if the tan is press-treated to remove moisture. Bulk for bulk, there are more heat units per unit of volume or of weight after a volume or weight of water has been expelled than there were when the tan was saturated and not pressed. If the fire is hot enough to dissociate the oxygen and hydrogen which compose the water, the heat for such dissociation is drawn from somewhere; doubtless from the flaming gases, where the process takes place, and, of course, they are cooled, and perhaps killed.

If such oxygen and hydrogen recombine, nothing is lost, and perhaps a mechanic-thermal advantage is reaped because the hydrogen flame is longer than the carbon flame. If, for any reason, such dissociated hydrogen does not get a chance to recombine from lack of temperature or time or room, there is a loss. Mr. Myers' results should serve to check the claims still advanced at intervals, that the combustion of steam gas is not a source of any great possible economy.

AUTHOR'S SUMMARY

In his discussion, Mr. Cary has described the McMurray tan furnace—one of many different types and designs now in use. All the ordinary forms of tan furnaces feed the fuel through holes in the top or arch over the grate. The number and arrangement of these feed holes vary in the different designs, but they all form a bed of fuel composed of cones of tan. For this reason they are all subject to the objection made by Mr. Cary, i.e., that the fuel burns away most rapidly around the bottom of these cones where the depth of fuel is least. The central parts of the cones offer great resistance to the draft so that active combustion takes place on only a small

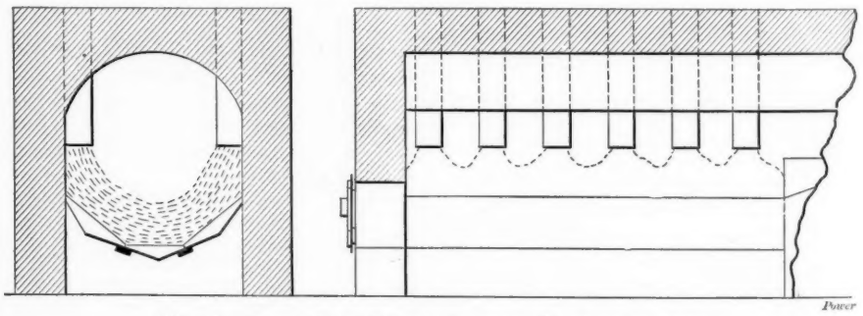


FIG. 9. BUSH TAN FURNACE WITH MULTITUBE FEED

percentage of the entire grate surface. This necessitates large grate surfaces and large furnaces with attendant radiation losses.

Another objection to the cone method of feeding the fuel, especially when only a single row of feed holes is employed, is that the fire is actually divided into a number of small fires around the bottom of the cones. This multiplicity of small fires separated by heaps of wet tan of low temperature results in lowering the furnace temperature and in retarding combustion.

In furnaces of this type with careless firing the writer has seen fully one-half of the grate surface doing no work at all in the way of any active combustion. These ill effects are best eliminated by very frequent feeding of the tan in small amounts, so that the percentage of wet tan in the furnace at any time is very small compared to the actively burning mass. High furnace temperature is thus maintained, more grate surface is active and the rate of combustion per square foot is greatly increased. The result is less grate surface required, smaller radiation loss due to smaller furnaces and greater ease in handling and cleaning the fires.

In general the greater the number of feed holes the higher will be the rate of combustion and the smaller the furnace required. Rapid firing in small amounts to equal the rate of combustion in the furnace is productive of best efficiency with any of the usual types of tan furnaces.

Tan presses of different makes, but all of the same type described by Mr. Cary, have been experimented with by the writer. It was found that with careful adjustment and attendance the presses would equal the performance quoted by Mr. Cary but that under tannery conditions of indifferent attendance and unskilled labor the presses do not maintain their efficiency.

The interference of the steam gas evolved from the fuel with the union of the combustible gases with the oxygen must be overcome by providing large combustion space, preferably over the fuel bed, by special baffles or by special draft action as in the writer's design of automatic furnace shown in Figs. 1 and 2.

The chemical composition of tan is assumed by Professor Kent to be practically the same as that given from an actual analysis in the author's paper. The heating value, according to Dulong's formula, is 7920 B.t.u. per pound, whereas the results of a large number of tests in a bomb calorimeter by Doctor Sherman show the heating value of a pound of dry hemlock tan to be close to an average of 9500 B.t.u.

The author desires to add that all results and data given in his paper are results of actual tests made under working conditions. No assumptions or theoretical calculations are involved in the conclusions. The feed water was in every case measured by means of two tanks or barrels set above a reservoir from which a separate feed pump supplied the boiler. Feed connections were so separated that it was physically impossible to pump the water elsewhere than in the boiler being tested. All connections involving a chance for leakage were blanked off. Valves were never assumed to be tight but were proved so during the entire test by means of an open-tee arrangement which would show any leakage.

The temperature of water entering as well as leaving the measuring barrels was taken at frequent regular intervals. The barrels were calibrated by weighing when filled to their overflow pipes with water at the temperature which the feed water had averaged during the test.

The fuel was in every case weighed in equal amounts to the fireman. A sample corresponding to each 200 pounds was taken, kept in closed receptacles and at the end of the test was mixed and quartered down to a quart or two-quart sample which was sent in sealed jars to Doctor Sherman for determination of B.t.u. and moisture. All readings and observations were obtained with like regard for accuracy of results.

Thermal efficiency is, of course, the safest and most accurate basis of comparing results of various boiler and furnace settings, and the highest result yet obtained in a reliable witnessed test in

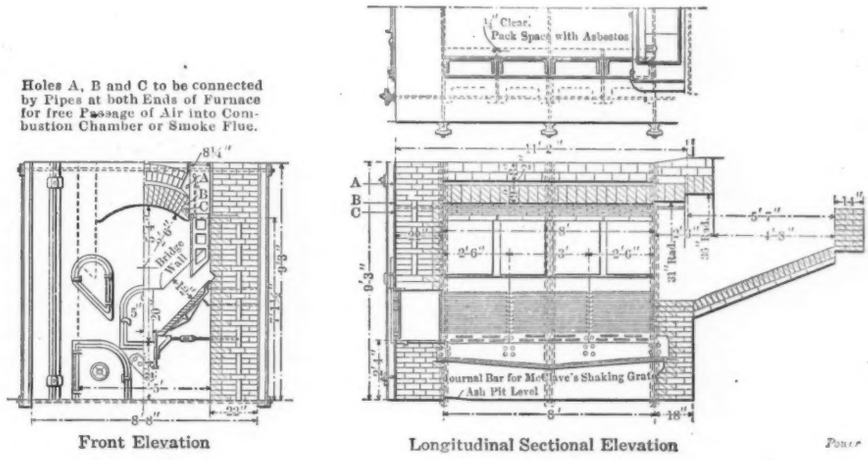

Holes A, B and C to be connected by Pipes at both Ends of Furnace for free Passage of Air into Combustion Chamber or Smoke Flue.
Front Elevation
Longitudinal Sectional Elevation

FIG. 10. MYERS FURNACE WITH ADJUSTABLE GRAVITY FEED

tan burning was 71.1 per cent. This is based on available heat in the fuel as fired after allowance is made for evaporating the moisture in the fuel. This test, which was made on the automatically stoked furnace before referred to, showed an efficiency of boiler and furnace of 54.4 per cent., based on the total heat of the fuel.

Power Value of Irish Bog Lands

In the Irish Year Book for 1909 appears an interesting article on the value of Irish bog lands for the generation of power. A short abstract follows:

The calorific value of 10 tons of peat, freshly dug and untreated, is assumed to equal that of one ton of ordinary coal. There are in Ireland 2,800,000 acres of unutilized bog land with a depth varying from 16 to 30 feet. There are 18,231 tons of peat to one acre of bog and this would be equivalent to 1823 tons of coal. Accordingly, the total calorific value of the bog lands of Ireland is estimated as being approximately equal to that of 5,104,000,000 tons of coal. Moreover, at least half of this quantity could be conservatively counted on as available ultimately for steam-raising purposes. Peat fuel equal to coal could be produced in selected localities in Ireland at $1.75 per ton. Some authorities maintain that the peat bogs are reproductive, and that in some cases the bogs increase in thickness 5 or 6 inches a year, according to the nature of the flora. This is simply theory, however, and there is doubt as to wherher they are recuperative at all. In case reproduction be at all feasible, the life of the bogs as a fuel reserve would, under adequate conservation, be capable of indefinite extension.

Economy in Generating Steam Power*

Saving Effected by the Use of Apparatus to Reclaim Waste Heat. Influence of Compounding Engines on Overall Economy

BY CHARLES EDWARD LUCKE†

It has required far more effort, and effort of a far higher order, to make the steam system of power generation as economical as it is today than was required to produce the first successful rotative engine. Without the study of efficiency and reduction of heat waste it is safe to say that but little progress would have been made in comparison with that which has been made. Even up to the year 1870, ninety years after Watt's successful engine, the use of steam-power systems in this country for stationary work was less extensive than the use of water power. The reason is to be found chiefly in the relative costs of power by the two systems up to that time. Although improvements in water-power machinery since then have decreased its cost somewhat and have made it possible to develop more difficult waterfalls, and electric transmission has increased the areas over which water power might be used, yet these advances are as nothing compared with the advances in steam-power systems. All advances in steam-power systems have come from analysis of losses through the methods of thermodynamics, a science which itself grew out of the earlier engines which furnished means for getting test data which thoughtful men compared, and in the comparison discovered the general laws.

Even with the best boilers and engines of the so-called simple type, the economy obtained is poor. The boilers give to the steam an average of 70 per cent. of the heat of the fuel or what is roughly equivalent, produce 10 pounds of steam per pound of good coal. The engines, even of the better sort, exhausting to the atmosphere use 30 or 40 pounds of steam per hour per horsepower, so that such boilers and engines together would require three to four pounds of coal per hour per horsepower, which value will rise in small and less well constructed units as high as six or eight pounds. A coal consumption of one pound per hour for each horsepower corresponds to an efficiency of approximately 17 per cent.; three pounds to about 5.6 per cent.; four pounds to about 4.2 per cent.; eight pounds to 2.1 per cent., so that when noncondensing steam plants with simple engines when burning from three to eight pounds of coal per hour per horsepower, which is the range they cover, are capable of converting into useful work only from 2 to 5 per cent. of the heat the coal contains. Improvements in the apparatus and methods of working have raised this value to about 15 per cent. in the best modern plants which seem extremely complicated compared with the simple noncondensing ones. Great as is this improvement, the efficiency still seems small, but it is difficult to realize the amount of time, material, money and brains that has been expended in making even this advance.

Such improvements as have been made are of two general classes. First, those belonging strictly to the engine to increase the work obtainable per pound of steam, and second, those relating to the efficiency of steam generation to increase the amount of steam made per pound of coal. Those which have been applied to the generation of steam have been directed toward the reduction of direct heat losses and the return to the boiler water of some of the waste heat. For example, while the hot gases leaving the boiler must be hot when they reach the chimney to produce a proper draft, they may carry more heat than is necessary for the draft, so that some heat is wasted. Also, the exhaust steam from the engine carries much heat away to be dissipated in the atmosphere, or given to condensing water. If, therefore, the cold water on the way to the boiler be brought in contact with the exhaust steam or flue gases, it will become warm, and therefore, require less of the heat direct from the coal to make it boil. As the heat for boiling the water is more and more derived from waste sources and less from the coal directly, the efficiency of steam generation will rise and more steam will be made per pound of coal. An apparatus designed to recover heat from the waste-flue gas is called an economizer and consists of a bank of tubes located in the flue or breeching and through which the water passes on its way to the boiler. To secure the advantage of heating the boiler feed water by exhaust steam feed-water heaters of two general classes are used. In one class the water passes through a bank of tubes set in a cast-iron casing through which the exhaust steam passes. As there is no physical contact between the steam and the water, the water remains pure. With feed-water heaters of the other class the water and steam mingle. Steam that condenses, together with any oil that it may have carried over from the engine, falls with the feed water to the bottom where filtering material is added to remove the oil. Feed-water heaters and economizers are only two of many means of returning to the boiler heat waste, that are now important subjects of study in plant economy.

As has already been explained, improperly controlled air supply to the furnace means heat waste of the most direct sort either by failure to burn all the fuel or by the use of too much air which carries away to the chimney too much heat. Much work is being done to reduce waste from this source and in this part of the problem perhaps the most important piece of apparatus developed is the mechanical stoker. Stokers are intended to maintain proper furnace condition continuously by feeding the coal mechanically to furnaces fitted with properly controlled air openings. The use of stokers is likewise associated with some saving in labor in handling the coal which is mechanically elevated from boats or cars to bins over the boiler room from which it runs down to each boiler through pipes.

These illustrations tell only the smallest fractions of the story of attempts to reduce heat waste and get more steam per pound of coal, but will serve to demonstrate the nature and direction of the efforts.

It is in the engines and their auxiliaries that the greatest advances have been made, but improvements directed toward the fullest utilization of the work capabilities of steam can be understood only when some of the important quantities are known. It is these quantities that show plainly the difficulties to be encountered in designing apparatus to carry out the processes to the fullest extent. It has already been explained that steam will do all the work it is capable of doing only when permitted to expand as much as it is capable of expanding in just the same way as a compressed spring may do work. It is important now to consider how much steam, under the ordinary conditions of initial and final pressure employed, may expand. A boiler pressure of 150 pounds per square inch is not considered high today, as pressures going up to 250 pounds are in common use, but even with the moderate pressure of 150 pounds to start with, one pound of steam would occupy a volume of 2¾ cubic feet. That is to say, one pound of water when turned into steam at 150 pounds pressure would occupy a volume of 2¾ cubic feet. If this steam, which may be likened to a com-

*Fourth Hewitt lecture delivered in Cooper Union hall, February 28.

†Professor of mechanical engineering in Columbia University.

pressed spring, be allowed to stretch until its pressure has come down to that of the atmosphere it will then occupy a volume of 26.8 cubic feet or it will have expanded 9.7 times, or, as it is often said, it will have suffered 9.7 expansions. If, however, it be allowed to continue to expand by the enlargement of the chamber in which it is working, until its pressure has fallen below atmosphere to within 97 per cent. of a perfect vacuum at sea level, such as is easily obtainable in our best modern condensers, then it would occupy a volume of 875 cubic feet or would have expanded 318 times its original volume. This means that one cubic foot of steam at 150 pounds pressure per square inch becomes 318 cubic feet when its pressure has been brought down to within 97 per cent. of a perfect vacuum. The more a spring is allowed to stretch the longer will it continue to push against whatever is restraining it and so the more work it will do. Similarly, the more the steam is allowed to stretch or expand the more work it can do. It appears, therefore, that steam expanding from this high pressure to the vacuum can do more work than if it expanded only to the atmosphere and it is interesting to note just how much more. The laws of thermodynamics indicate that when expanding from the high pressure to atmosphere, steam can only do about 54 per cent. of the work it could have done had it expanded to the vacuum.

About the smallest clearance volume, that which even the best arranged valves will permit, may be set at 2 per cent. of the cylinder volume, and even this is almost dangerously small. To carry out such expansion as has been discussed, the volume of steam at the beginning of the expansion would be only about 0.3 of 1 per cent. of the volume at the end of expansion, or less than the dead clearance space itself. If steam were admitted to the cylinder for only 1/20 of the stroke of the piston and then cut off it could expand only 14 times. If steam were admitted only to fill up the waste space and did not follow the piston in its movement at all, then it might expand only 50 times instead of the 318 times, necessary for it to do all the work it is capable of doing. These figures show the absolute impossibility of carrying out this expansion process in a single cylinder and why single-cylinder or simple engines are necessarily wasteful.

When, however, the work is done in two cylinders in succession or even three or four, the conditions are changed. This process of successively working the steam in several cylinders in series will permit of more expansion in such cylinders as we can construct than is possible in a single cylinder. Suppose, for example, the first of the series to be just large enough to hold the pound of steam when full, then if the largest cylinder has a volume 318 times this, the complete expansion would be possible. This would call for a low-pressure cylinder diameter about 15 times that of the first high-pressure cylinder. These so-called multiple-expansion engines are, therefore, capable of making better use of the steam than the simple engine.

When several cylinders use the steam in succession, permitting the pressure to fall in steps and producing the work of the steam in stages, the cost of the engine increases. A two-cylinder or compound engine will cost about 50 per cent. more than a simple single-cylinder engine, while third and fourth cylinders for triple and quadruple expansion each add still more initial expense which is the price to be paid for increased steam and coal economy. Evidently a limit will be reached somewhere beyond which successive expansion cylinders will cost more than the fuel saving will warrant. This limit is found with the two-stage or compound engine in stationary and the three-stage or triple engine in marine practice.

After all the improvements of engine, boiler and auxiliary equipment and many others of minor importance, each of which is the result of a great amount of patient study, there is yet required on the part of the engineer good judgment to decide whether the expense incurred in saving heat will be warranted by the saving it will effect in heat or coal.

A comparatively recent analysis of the distribution of coal energy in a large central power station about as highly efficient as any is that made by H. G. Stott, superintendent of motive power for the Interborough Railroad Company, quoted below and offered to show just where the heat goes.

ANALYSIS OF THE AVERAGE LOSSES IN THE CONVERSION OF ONE POUND OF COAL INTO ELECTRICITY.

	Per Cent.	Per Cent.
1. B.t.u. per pound coal supplied	100.0	
2. Loss in ashes		2.4
3. Loss to stack		22.7
4. Loss in boiler radiation and leakage		8.0
5. Returned by feed-water heater	3.1	
6. Returned by economizer	6.8	
7. Loss in pipe radiation		0.2
8. Delivered to circulator		1.6
9. Delivered to feed pump		1.4
10. Loss in leakage and high-pressure drips		1.1
11. Delivered to small auxiliaries		0.4
12. Heating building		0.2
13. Loss in engine friction		0.8
14. Electrical losses		0.3
15. Engine radiation losses		0.2
16. Rejected to condenser		60.1
17. To house auxiliaries		0.2
	109.9	99.6
Balance		10.3

From this it appears that 60 per cent. of the heat of the coal is finally discharged in the condensing water and nearly 23 per cent. to the stack for the maintenance of draft, these two together aggregating about 83 per cent. of the heat in the coal. No way has ever been found to reduce this heat that must be delivered to the condensing water. It is an essential characteristic of the steam system and the fact that steam in condensing, at however low a pressure, does give up to the condensing water so large an amount of heat, constitutes a final limit of efficiency inherent to this system. The stack loss can be entirely eliminated, but this elimination would mean the substitution of some other means of producing draft and no means has as yet been found that is as cheap as the loss itself.

An Ice Factory in Smyrna

The modernizing effect throughout Asia Minor of the establishment of an ice factory at Smyrna is revealed in the following report from Consul-General Ernest L. Harris; which appeared in the *Monthly Consular and Trade Reports:*

An American company with a capital of $160,000 erected a large plant here last summer capable of producing 70 tons of crystal ice daily. It was believed by many that the new plant would soon close down owing to the limited demand, but depots were established at various points of the city, whence the ice was delivered in small push carts to all outlying districts, with the immediate result that the daily demand rose to over 50 tons.

The company is now installing refrigerators on the hire system, and it is expected that the utmost capacity of the plant will be taxed in order to meet the general demand for ice among those who previously were unacquainted with its use or lived too far away from the old plant to be able to obtain a regular supply.

The erection of this ice plant in Smyrna is having its effect on all the other large cities in Asia Minor and the islands of the archipelago. Within a year or so they will all be supplied with such plants, and the machinery is almost certain to be American. This office is continually receiving inquiries in regard to the cost of such enterprises, and the ice plant here is being visited by tourists as one of the attractions of the city.

Power and The Engineer
Devoted to the Generation and Transmission of Power

Just a Business Proposition

Two years ago a committee of Senators and C ngressmen spent some months investigating the postal situation. They took nothing for granted but looked into every nook and cranny that was open.

This commission decided that no reliable data were available and that most of the figures given were estimates."

They found that very few improvements in the business methods of the Post Office Department had been made in a hundred years.

They found the largest business in the world run with antiquated methods, that the responsible positions in this vast machine were in the hands of politicians, that red tape clogged the wheels of progress and that, unlike any other business concern, they were trying to economize by cutting down the volume of their business rather than increasing it.

This committee offered a bill to Congress, known as the Overstreet bill—but Congress had other fish to fry and no action was taken.

This bill provided for the appointment by the President of a nonpolitical Director of Posts, who shall not be removed except for cause, and for a reorganization of the Postal Department along modern business lines.

Senator Carter has reintroduced this bill, and it ought to pass.

It makes no snap judgments on "estimates." It provides that the department be put on a business footing first and remedies applied after it is known what is the matter with it.

The increase of magazine postage is not needed—the deficit increased $18,000,000 on a *decreasing* second-class mail.

There is something rotten in Denmark—we think it is just departmental dry rot.

There have been some seventeen Postmasters General in the past twenty years. To change the head of any big private business concern that often would break it. Fortunately the Postmaster General just rattles around in his job and the clerks carry on the work and make out their reports—and his.

Then he proposes—but the people dispose.

And the people of this country want their newspapers and magazines just as cheap as they can get them and they are getting tired of some things in our antiquated post office.

For instance, they are wondering why our department can make arrangements to carry a parcel from Seattle via New York and on to London or Paris or Berlin for 12 cents a pound *and charge 16 cents a pound to carry the same package to Tacoma, or even across the street.*

They are tired of the trade our post office made with Canada increasing the rates on second-class matter to our neighboring country to the detriment of our trade there. They are tired of such acts as the recent reduction of 80 *per cent.* on foreign-going letters—and a howl about a deficit in the same breath.

If parcels posts are good for the foreigner they are good for us.

If the postage on the letter that carries the money order back to Italy should be less, why should the postage on the paper that tries to make an American citizen of another Italian be more?

A respectful letter from any business man to his representative in Congress asking him to support the Carter Bill (Senate Bill 6287) will receive attention.

Let's look over the books before we shoot somebody for embezzlement.

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Necessity of a Better Tube Joint

The new twenty thousand-ton, twenty-one-knot battleship "Delaware" is equipped with fourteen water-tube boilers designed for a pressure of two hundred and ninety-five pounds. The tubes in those boilers are 0.203 inch thick. This corresponds to No. 6 Birmingham wire gage. The thickness of a standard four-inch tube is 0.134 inch, or No. 10 Birmingham wire gage.

Why, then, did the Government use tubes four gages thicker than standard when the latter were amply strong for the pressure required? To get a better tube joint; and the recognition of this necessity, and the means adopted to secure the greater strength, are worthy of comment.

The tube joint has always been a great problem with the fire-tube boiler, and many are the methods and expedients that have been used to produce satisfactory results. It is as great a problem with the water-tube boiler, though less apparent. Many engineers overlook the fact that a tube in a water-tube boiler is subjected to a strain equal to its internal area times the internal pressure, tending to pull it out of its tube sheet. It is not so much a question of rupturing the tube longitudinally as it is a question of the strength of the joint made by the tube in the tube sheet. In the case be-

fore us, there is a strain of about one and one-half tons, tending to pull each four-inch tube in the "Delaware's" boilers out of the tube sheet. Nor is it altogether a question of keeping the tube from being pulled out, but, rather, that the tube shall not slip in the tube sheet and thus leak.

There are many ways of increasing the holding power of the tube: by harder rolling; by flaring; by double taper; roughing the hole in the tube plate; or by such a method as, for example, the Caramen, in which a couple of grooves are cut in the tube sheet opposite similar grooves in the tube end and rings of wire placed in the grooves; but the first three give little increase; the fourth is questionable, and the last is too expensive. The best all-around method as regards cost, durability, facility and ease with which an ordinary mechanic may remove and replace a tube, is to use thicker tube ends which, of course, means thicker tubes.

It is of further interest to note that the Government is not afraid that the use of thicker tubes will diminish the rate of transmission of heat or the efficiency of the boiler. They know that the slight increase in thickness will have but little effect in that respect.

Horizontal versus Vertical Gas Engines

There is a noticeable difference of opinion among builders in this country as to the relative advantages of the horizontal and vertical types of gas engine. Minimum wear on cylinders and simplicity of cylinder and piston details have always been features of the vertical type, and greater accessibility of the whole engine and ease of adjustment of the main bearings have been prominent amongst the claims urged in favor of the other type. Many other considerations come into the problem, but there can be no question that each type embodies certain advantages, leaving the manufacturer the choice of developing the type which, in his estimation, embodies the least number of disadvantages.

In visiting one factory it will be found that the manufacturer is wedded to the vertical type, claiming for it better economy and a lot of other desirable features, while in another factory horizontal engines are considered the only type worth building. This healthy difference of opinion is fortunate, as it tends toward the most rapid development of both types. In the design of the horizontal engine the influence of German practice is easily traced, and there is reflected in the larger, and lately even in the smaller, engines of the horizontal type a tendency to follow more closely than ever the best German practice, which certainly should have an influence for reliability on long, continuous runs and reasonable maintenance expense, these being the only points upon which the gas engine is questioned today.

In England, where the high-speed vertical steam engine has reached its highest state of development, there was at first a hesitancy to adopt the vertical gas engine owing, no doubt, to the fact that the overwhelming majority of Continental European makers build horizontal engines. Of late, however, there seems to have been a notable increase in the number of English vertical gas engines. Nearly all builders now supply this type in medium powers, and it is reasonable to expect that, with the experience in vertical steam-engine development to draw upon, this type of gas engine will reach a high state of perfection in England.

In large sizes the horizontal engine has held the field everywhere, and in discussing the tendency of the gas-engine industry the assertion has been frequently heard that either vertical or horizontal engines might be safely installed in sizes up to four or five hundred horsepower, but for anything over the latter figure only horizontal engines should be considered. Changes take place in the engineering world, however, as rapidly as in any other branch of human activity, and the question can hardly be so easily disposed of at the present time. Vertical double-acting marine gas engines of five hundred horsepower have been built in this country for some time and there are now in service here six hundred horsepower vertical single-acting stationary engines, while in England there have been built several four-cylinder tandem vertical gas engines of one thousand horsepower, and there is every reason to believe that the end is not yet.

Any forecast of the probable trend in gas-engine design would naturally be based on the past development not only of the gas engine but of the steam engine, and it is reasonable to look for a gradual development of the internal-combustion engine along lines similar to the development which has characterized the steam engine. In fact, this tendency can already be traced to some extent in the latest designs. However, the process of adapting the internal-combustion engine to the various classes of service will, of itself, work out the problem of horizontal versus vertical types, but there is no reason to doubt that we will continue to find both types in the field, each working under the conditions for which it is peculiarly fitted.

Disastrous boiler explosions have been happening of late with alarming frequency—and the shame of it is that they are nearly all preventable.

Selling Second-hand Boilers

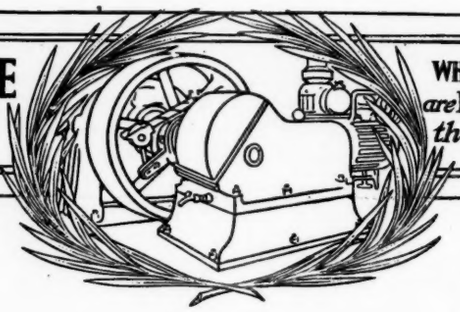
Engineers are concerned more or less with the elastic properties of materials. From gage glasses possessing little elasticity to rubber packing rings possessing a great deal, the range is very considerable. The elastic qualities of many of the materials and things that lie within these limits are interesting, but none is more inexplicable or harder to understand than the elasticity of the conscience of the average steam user when he wishes to sell a second-hand boiler. He may be a pillar in the community, and a most conscientious man in all other business transactions, but when it comes to disposing of a second-hand boiler, he seems to take, as it were, a moral holiday, and lets what is unscrupulous in him, and hidden from sight under ordinary conditions, run riot until the boiler is sold.

We would expect such a man to say: "Here is a boiler which has been condemned, and because it is not good enough for me, it is not good enough for you, and it is valuable only as scrap, and I will accept no greater amount than that which its weight justifies, and I have further caused the words 'Condemned' to be stenciled in the front head, so that no unsuspecting person may be led to use this boiler as a steam generator, and thus endanger his life and property and the lives of others." But does he do this? No, he gives the boiler a nice coat of thick tar paint and chuckles at his shrewdness as he palms it off on an unsuspecting public.

Just as all horses offered for sale are three years old, so all second-hand boilers are offered for sale because the plant has outgrown them, and they are now too small for the work. Possibly, this may be the case, and it is possible that the thick coat of tar paint which is invariably given the boiler when it is offered for sale may be in the nature of an affectionate testimonial on the part of the owner in behalf of the good service the boiler has rendered in the past; it may be merely a natural pride in having the object which he offers for sale appear at its best, but we doubt it.

If a factory closes down permanently, and the whole plant is sold piece-meal, then one would be justified in assuming that the boilers offered for sale were in as good condition as the length of time they had been in service would warrant, and in such a case one can expect to pay a reasonable price and not to get a great bargain, but it is seldom that a large plant is abandoned and the machinery sold in this manner, and generally when a second-hand boiler is offered for sale by a concern still in operation, it is because that boiler is no good.

NEW POWER HOUSE EQUIPMENT



WHAT *the* INVENTOR *and the* MANUFACTURER *are* DOING *to* SAVE TIME *and* MONEY *in the* ENGINE ROOM *and* POWER HOUSE

A Novel Integrating Indicator

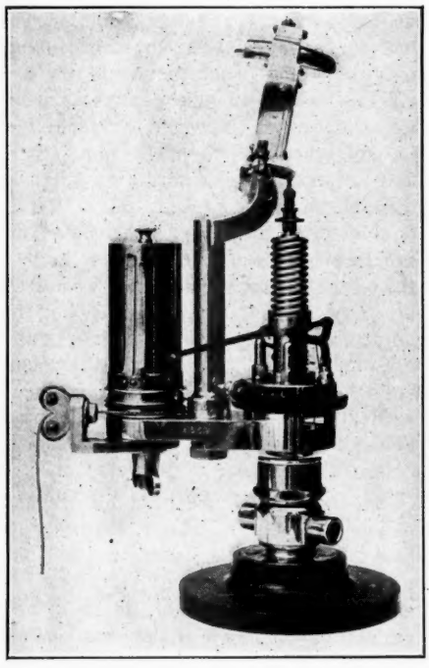
By Alfred Gradenwitz

The use of the ordinary indicator, which only allows a single diagram, out of some hundred developed by the machine, to be chosen for ascertaining the average indicated output during the testing period (5 to 10 minutes or more) is undoubtedly inadequate. While giving rise to many errors, even in connection with engines the load of which is properly balanced, it is apt to fail entirely in the case of those engines which in normal operation work under continual variable loads, affording no possibility of balancing. This shows how urgent a want is felt for an apparatus allowing the average indicated output of a reciprocating engine to be ascertained by taking into account each of the diagrams developed. Endeavors have therefore been made from time to time to introduce an apparatus of this kind into general practice. The apparatus designed by W. G. Little and Ch. Hamann, respectively should, for instance, be mentioned in this connection, though no reliable information as to their performance in permanent operation is available. The same applies to the totalizing indicator, so-called, by Hlawatscheck, a description of which has been published without any experimental results in a book on "Technical Investigation Methods," published by Julius Brand, of Berlin.

The serious difficulties to be overcome in designing an apparatus of this kind can be inferred from the following opinion expressed by Professor Frese on the Hamann apparatus: "The mass effects of the recording device are liable to produce considerable errors; the necessity of throwing the apparatus in and out at the beginning and end of the stroke respectively is felt a great inconvenience, while requiring an excessive number of records for compensating errors in the case of omissions. The friction on the parchment tape will be lost rapidly, and the increase in smoothness even shows the wearing of the traveling roller; the small disks do not lend themselves for easy reading."

An apparatus recently constructed on plans by A. Boettcher at the works of H. Maihak, Hamburg, allows an elliptical diagram of given area to be inscribed with variable numbers of turns, its area being counted by the apparatus. Comprehensive tests have shown this apparatus to be exceedingly practical for ascertaining the actual average indicated output of a reciprocating engine. By immediate reading off a counter, and multiplying with a given constant, the average indicated pressure is ascertained for any desired period of observation, and the apparatus, even in the event of the engine load being subject to continual violent fluctuations, takes account of each one of the diagrams developed. This integrating indicator is arranged as follows:

The piston rod of the indicator is made to displace in an axial direction by means



Boettcher Integrating Indicator

of an angle lever, a roller unwinding from the upper front surface of the indicator. This axial displacement of the roller is proportional to the displacement of the indicator piston, and accordingly, to the instantaneous value of the indicated pressure in the working cylinder. The arc unwound from the roller thus becomes proportional to this pressure. On the other hand it is proportional to the path of the piston, as the roller is operated from the indicator drum. The finite arc unwound from the roller thus becomes proportional to the sum of the products of pressures by the elements of the piston path. The axis of the roller operates through a worm and counter, allowing outputs of up to ten working hours and upward to be recorded permanently. The apparatus is cheap, and can be fitted to any existing indicator, which may then be used at will with or without the integrating (summing) device.

Actual experience has shown the adhesion pressure of the roller to be quite sufficient for an accurate record up to the maximum speeds usual in connection with steam engines and motors. The indicator springs can be so designed as to keep their strain within sufficiently low limits to exclude any deformation of the scale. Theoretical calculations as to the possible errors due to the mass effects of the indicator piston, rods and counter led to so small figures that these factors can be left entirely out of account. In a similar manner the friction resistances of the counter can be shown to influence in no appreciable degree the accuracy of results. In fact, even in the case of permanent operation, readings can be taken with sufficient accuracy (to 1 per cent. nearly).

A recent improvement of the apparatus consists of fitting it with electrical contacts, allowing readings to be taken at any place desired, even of any number of indicators at a time.

In connection with a demonstration recently made before the Hamburg section of the Association of German Engineers, Messrs. Ph. Holzmann, who are engaged in building the Elbe tunnel, placed one of their air compressors showing very constant diagrams, at the disposal of the inventor. The engine was run at slightly above 100 revolutions per minute, the diagrams being recorded and counter readings being taken on the high- and low-pressure steam cylinders as well as on the air cylinders. The results of these tests are extremely advantageous; the original diagrams recorded with or without the counter on the same sheet of paper distinctly show by projection on a screen, the nearly perfect coincidence of the two diagrams. In fact, apart from some slight mass oscillations shown by the counter diagram at the beginning of the stroke, but which would soon be damped, the two curves coincided completely. The readings of the counter also gave every satisfaction. These measurements were carried out in the following manner: During the counting, which lasted an hour, a

diagram was inscribed every two minutes without exchanging the sheet, so as to obtain a number of diagrams, the maximum and minimum areas of which could be ascertained by means of a planimeter. Owing to the very slight departures between the various diagrams in that special engine, the reading of the counter then must fall between these two figures. One maximum-area diagram showed an average hight of 0.562 inch and a minimum-area diagram a hight of 0.54 inch; the counter which during the experimental period had recorded 3090 individual diagrams, showed the average hight of the diagrams to be 0.556 inch, which is intermediary between the two figures above given. Another apparatus recorded likewise during half an hour an average hight of 0.54 inch, while the maximum and minimum diagram areas yielded 0.542 and 0.536 inch respectively. Similar results were obtained in connection with other engines running at 80 and 100 revolutions per minute.

The apparatus has recently been further improved by the addition of an equilibrating spring, so as to make it independent of any vibrations or shocks, and to fit it also for a horizontal arrangement; in this improved shape it is suited for use in connection with engines aboard vessels.

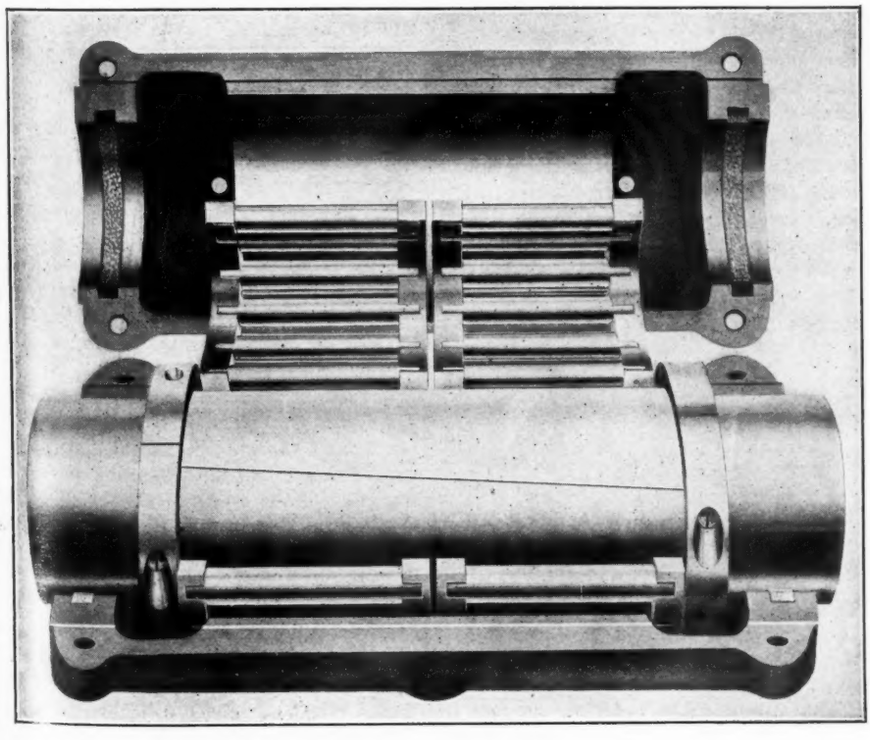
New Sells Roller Bearing

The Royersford Foundry and Machine Company, Inc., of Philadelphia, Penn., has recently put on the market a new roller bearing designed by John D. Sells, who for years has been identified with the Standard roller bearing. In the accompanying illustration is shown one of these bearings which it is claimed is universal in adaptation. Any bearing of a given size can be fitted to any hanger, post-hanger or pillow block of a corresponding size, either for a new installation or for substituting in a plant previously equipped with other bearings. The bearings are split so that it is an easy matter to apply them. From the illustration it will be apparent that the bearing has a self-contained roller structure—a split steel cage holding the case-hardened steel rolls equally separated from each other, which insures that the rolls will revolve parallel with one another.

To prevent the shaft from being cut and scored by the case-hardened steel rolls, the bearing is fitted with a hard-steel split bushing, clamped on the shaft by two split collars, and in order to increase the adaptability of the bearings, these bushings can be varied in thickness, so that the bearings may be fitted to different diameters of shaft. Each roller structure is susceptible to three such changes of bushings. The steel split collars which clamp this bushing to the shaft also perform another very important duty. Being fast to the shaft they rotate at a speed three times as great as that of the roller structure itself, and therefore provide a rotating surface for the end of the roller structure to run against, thus aiding the latter considerably.

The two halves of the bearings are fitted together by milled joints in order to make them dust proof, and to farther insure this point, a heavy felt wiper is placed at each end of the box. This also serves the purpose of preventing oil from running out of the box, which is fitted with drain holes so that it may be flushed with kerosene oil or other cleansing fluid when the oil thickens or gums. Tapered holes at the top of the box provide for oil cups.

For head-shaft or jack-shaft purposes $3\frac{11}{16}$ inches or larger, the manufacturer equips the bearing with a double roller structure so as to provide a longer and more suitable bearing surface for the heavily loaded shafts.



SELLS ROLLER BEARING

Annual Meeting of Wisconsin Engineers

A. A. Babcock, Jr., chief engineer of the Wisconsin Valley Improvement Company, discussed storage reservoirs for water power, in the State, at the second annual meeting of the Engineering Society, of Wisconsin, held at Milwaukee, February 23, 24 and 25, beginning with the first agitation of the subject about 1879, at the time the general Government started its work of creating storage reservoirs on the headwaters of the Mississippi to improve the navigation of that river. He traced the changes made in the bill introduced in the legislature of 1907 for the incorporation of the company, showing that those interested welcomed every change which could safeguard the public and pointing out the results already achieved. The company had expended approximately $140,000, and in the month of February, usually one of very low water in the river, had a fairly good head at all powers. It now has sixteen storage reservoirs and the total power will ultimately be tripled.

The present status of hydroelectric power transmission in Wisconsin, particularly with reference to switchboard facilities and line equipment, was discussed by M. H. Collbohm and C. V. Seastom, of Madison.

Officers for the ensuing year are C. V. Kerch, of Janesville, president; W. C. Lounsbury, of Superior, vice-president; A. A. Babcock, of Wausau, and Prof. G. I. D. Mock, of Madison, trustees, and W. G. Kirchoffer, of Madison, secretary.

According to the *Mechanical World* four parts alabaster plaster and one part of finely pulverized gum arabic mixed with a cold-saturated borax solution into a thick paste, make an unequaled all-round cement for stone, glass, bone, horn, porcelain and wood, which becomes hard as marble and possesses the agreeable quality of not solidifying immediately after mixing, but only after from 24 to 30 hours.

INQUIRIES of General Interest

QUESTIONS are NOT ANSWERED UNLESS they are ACCOMPANIED by the NAME and ADDRESS of the INQUIRER.

Capacity of Ammonia Compressor

How do you determine the refrigerating capacity of an ammonia compressor, 10x16-inch, running at 75 revolutions per minute? How many feet of 2-inch pipe can one allow for refrigeration and how many feet of 2-inch pipe for ice making?

A. R. M.

A double-acting compressor having a stroke of 16 inches and running 75 revolutions per minute has a piston speed of

$$\frac{2 \times 16 \times 75}{12} = 200$$

feet per minute.

The area of a 10-inch piston is 78.540 square inches = 0.5454 square foot. The volume swept through by the piston under the above conditions is the same as though it traveled in one direction and would be

$$0.5454 \times 200 = 109.08$$

cubic feet per minute.

There is no generally adopted standard among builders of refrigerating machines for fixing the rating of compressors. The number of cubic feet of piston displacement per minute to produce a cooling effect equal to the melting of a ton of ice varies widely with the condition of liquid temperature at the expansion valve and vapor pressure within the expansion coils, and, of course, varies with the efficiency of the compressor or the ratio of the actual to apparent displacement.

Assuming the compressor has 100 per cent. efficiency, 8 cubic feet would be required under the conditions of 200 pounds head pressure and 0 pound back pressure; while at 100 head pressure and 40 pounds back pressure only 2.05 cubic feet would be required per minute per ton of cooling effect per twenty-four hours.

In a paper read before the American Society of Mechanical Engineers, June, 1905, on the subject of Standard Unit of Refrigeration, it was determined that the average practice of large builders corresponds very closely to 4.4 cubic feet per minute from liquid at 90 degrees Fahrenheit and evaporating at 0 degree Fahrenheit or 15.67 pounds back pressure, assuming a compressor efficiency of 90 per cent. There use of the factor of 4.4 would give

$$\frac{109.8}{4.4} = 24.7 \text{ } tons.$$

For direct-expansion ammonia piping in rooms of ordinary temperature (36 degrees) 350 feet of 2-inch pipe should be installed per ton. This corresponds to a difference of temperature of 22 degrees Fahrenheit between the ammonia on the inside and the air on the outside of the pipe. Where a greater difference exists a smaller amount of pipe surface can be used and when the difference is smaller more surface is required. In ice making by the can system 90 to 100 square feet of pipe surface is used.

How Long Will Engine Run?

If a tank having a capacity of 112,000 cubic inches is given one charge of 300 pounds air pressure, how long will an engine run before coming to a stop; the engine to develop an average of 4 horsepower at 20 pounds pressure, consuming 216 cubic inches per revolution?

H. O.

If the engine is supplied with compressed air at 20 pounds pressure at the throttle, it may be reasonably assumed that the mean effective pressure in the cylinder will be about 14 pounds. With this mean effective pressure the engine will have to run at a speed of about 530 revolutions per minute in order to develop 4 horsepower if the piston displacement per revolution is 216 cubic inches. Assuming further that the release pressure of the air in the cylinder at the end of the stroke is 10 pounds above atmosphere, the engine will consume about 120 cubic feet of free air per minute. This allows for loss in the clearance space and slight piston leakage. If the tank having a capacity of 112,000 cubic inches is charged with air at 300 pounds pressure, it will contain:

$$\frac{112,000}{1728} \times \frac{314.7}{14.7} = 1390$$

cubic feet of free air.

As it is assumed that the engine is to use air at 20 pounds pressure it will be possible to allow the pressure in the storage tank to be expanded down to 20 pounds before the engine will stop. When the tank is under 20 pounds pressure it will contain:

$$\frac{112,000}{1728} \times \frac{34.7}{14.7} = 153$$

cubic feet of free air.

The difference between the amount of air in the tank when charged at 300 pounds pressure and when charged at 20 pounds pressure will be the amount of air available to run the engine. This amount will be 1237 cubic feet of free air.

As it has been calculated that the engine will use about 120 cubic feet of free air per minute, it will consume 1237 cubic feet of free air in about 10 minutes. The engine would not come to a dead stop in this time, but the speed would gradually fall off until stalled by the machinery it was driving.

High Speed Engines

Who made the first high-speed and shaft-governed engines?

F. R. L.

A Mr. Hartnell, in England, was, so far as is known, the first to use the shaft governor in steam-engine construction, which he patented about 1860. In 1862, Charles T. Porter exhibited in London his high-speed engine, which, however, did not have a shaft governor. Previous to this J. C. Hoadley and Harris Tabor had each made centrifugal or shaft-governed engines. John E. Sweet built the first straight-line engine in 1871. It is probable that Armington & Sims, of Providence, R. I., in the early eighties were the first to make a business of supplying shaft-governed, high-speed engines.

Temperature of Steam

Is there any way of determining the temperature of steam from its pressure other than by reference to steam tables?

F. R. D.

The approximate temperature may be found by multiplying the sixth root of the absolute pressure by 200 and subtracting 101 from the product. The sixth root of a number may be found by extracting the square root of its cube root.

Superheated Steam

If the steam from one of two boilers is superheated 100 degrees and that from the other is not, will the steam in the header into which the boilers discharge be superheated?

J. F.

There would be superheated steam in the header, provided the steam from boiler No. 1 did not lose all its superheat by radiation before it got there, and that the steam from No. 2 was not so wet as to soak up all the superheat.

NEW PUBLICATIONS

THE DESIGN AND CONSTRUCTION OF INTERNAL COMBUSTION ENGINES. By Hugo Güldner. Translated from the German by H. Diederichs. Published by D. Van Nostrand Company, New York, 1910. Cloth; 700 pages, 8½ x11 inches; 728 iliustrations; 36 folding plates; 151 tables. Price, $10.

Güldner's original work has been regarded in Europe as a standard authoritative reference book for designers and builders of internal-combustion engines ever since the first edition was published about seven years ago. The present translated edition, due to the remarkable energy and high degree of ability and good judgment of Professor Diederichs, will undoubtedly be awarded the corresponding status here. It is a monumental work —almost encyclopedic in scope and treatment.

The work, though complete in one volume, is divided into four parts, and these are subdivided into sections which, in turn, are divided into specific subjects. Part I contains three sections: General Considerations, The Cycles, and Critical Examinations of Cyclic Events. The general considerations comprise a classification of engines by cycles and cylinder characteristics (it is with regret that we note Professor Diederichs' use of the slovenly terms "two-cycle" and "four-cycle"), and a presentation of thermodynamic definitions. In the course of the latter subdivision, the author takes issue with Mallard and Le Chatelier as to the specific heat of gases at different temperatures, and cites other authorities in support of his contention that the specific heat may be considered as remaining constant within the temperature and pressure ranges of gas-engine practice. The second section presents a mathematical analysis of the "constant volume" or Beau de Rochas cycle and the "constant pressure" cycle of Brayton, Hargreaves and other investigators, and as only the two cycles are considered, the section title "Various Cycles of Operation" is a little broad. In this section the author gives the thermodynamic characteristics of the cycles and practical considerations of them, and follows the common habit of classifying the Diesel cycle as "constant pressure," which it is not, of course, in actual practice, though intended and expected to be originally. Section C of Part I is devoted to mathematical analysis of each event of each of the cycles followed out in commercial engines, and practical considerations of them. As the book is intended for designers, the table on page 32 might have been elaborated with propriety to give compression pressures and temperatures for a greater number of practical conditions. As it is, only one precompression pressure is considered, four values of the exponent n of the curve equation, and twelve compression ratios. However, the charts on pages 33 and 35 fill out many of the intermediate values omitted from the table, and most designers prepare their own charts or tables to fit the particular conditions applying to their own practice. A comparison of the two-stroke and four-stroke cycles as carried out in actual engines closes Part 1, and it is the most clear-sighted sensible discussion of the subject that the reviewer has ever read.

Part II is devoted to the Design and Construction of Internal Motors [engines]. Section 1 is a plea for "less invention and more rational design," which is thoroughly justified but will probably fail in effectiveness, because the half-baked designers who turn out freaks do not read books of the depth of this one. The other sections of this part are devoted to discussion of the merits of horizontal and vertical engines, single-acting and double-acting cylinders, different numbers of cylinders, compounding, ratio of stroke to diameter, determination of principal dimensions,. general engine parts, methods of governing, ignition systems, etc., etc., This portion of the book is much the best of its contents. It is authoritative without being pedantic and adequate without being prolix.

Part III relates to the building, erection and testing of engines, and contains some data on costs that are not usually found in print. It also contains descriptions of some representative Belgian, German and Austrian engines, with reports of performance tests and very complete operating data, and unusually detailed descriptions of the principal American engines. Automobile and other portable or self-propelled types are given considerable attention also.

Part IV comprises a very full discussion of fuels and combustion phenomena, and is followed by an appendix containing a simplified exposition of the theory of gas-engine thermodynamics, some points taken from operating practice, the general specifications for the purchase of machinery adopted by the Verein Deutscher Maschinen-Fabriken, and various codes of rules and regulations for the installation, operation and testing of engines and producers.

It is to be regretted that the translator did not revise the author's application of symbols more thoroughly, at least to the extent of avoiding the use of the same letter to express more than one meaning. A complete list of the symbols and their meanings should have been provided, too, for a work as elaborate and comprehensive as this, preferably at the beginning or close of each general division of the subject matter.

Publications issued by the United States Government during the past month, which will be interesting to our readers, are:

Psychrometric Tables for obtaining vapor pressure, relative humidity and temperature of dew point from readings of wet and dry bulb thermometers, by C. S. Marvin; publication 235; price 10 cents.

Annual report of Supervising Inspector General of the Steamboat Inspection Service, containing the revised rules of the steamboat inspectors, to be had upon application to the board.

A list of documents upon fuel testing and results of coals and other fuels made by the U. S. Geological Survey, to be had by addressing the Geological Survey.

Gas Producers. Recent development of producer gas power plants in the United States, with list of the Survey's publications on fuel testing, by Robert H. Fernald; price 15 cents.

Production of natural gas, by B. S. Hill; price 5 cents.

Production of Peat in 1908, by Chas, A. Davis; price 5 cents.

Coals and Laws and Regulations thereunder, with amendments and supplemental circulars; price 5 cents.

When not otherwise specified these publications can be had from the Superintendent of Public Documents, Washington, D. C.

McGraw-Hill Book Company, 239 West Thirty-ninth street, New York City, has just issued a new descriptive catalog of technical books. The catalog is divided into three parts, the first dealing with electrical books, Part 2 with books pertaining to mining and metallurgical subjects, and Part 3 with books on civil and mechanical engineering; steam, gas and water power; mathematics and miscellaneous. Each part, or all three parts, as desired, are sent free on request.

OBITUARY

F. K. Rittenhouse, the former blind member of the board of examining engineers, of Chicago, who last May resigned following an investigation of the board, died Monday, February 21. Mr. Rittenhouse was born in Camden, Ont., forty-three years ago and had lived in Chicago for the past thirty years, entering first the employ of the Rittenhouse and Embree Lumber Company as engineer, where in overexerting himself a blood vessel in his head burst, and blinded him. This was in August, 1904, and the accident made him unfit for his duties. He resigned and entered other business, but by constant study and ability to apply what he had learned he soon became almost as efficient an engineer without sight as he had been with it, and in July, 1907, he secured the appointment to membership on the board of examining engineers. He held the position almost two years, when the investi-

gation was directed against other members of the board. Though at the time it was affirmed that the blind engineer was in no way responsible for the alleged irregularities in the office, it was said that owing to his blindness he was unable to observe all that was happening, and on this ground his resignation was requested. The cause of death according to the physician's diagnosis was cancer, but it is said that worry over his enforced resignation had much to do with the hastening of the end. Mr. Rittenhouse is survived by his wife and one son, Frank Rittenhouse, Jr.

SOCIETY NOTES

The mechanical display at the next annual convention of the American Order of Steam Engineers will be a record breaker, judging from the rapidity with which the applications for exhibition space are being received. The convention will be held at Odd Fellows Temple, Philadelphia, Penn., June 6 to 10. Fred L. Jahn, of the Watson & McDaniel Company, Philadelphia, is the secretary of the American Supply Men's Association, which has the exhibition in charge.

Among the pleasant features arranged for the winter's educational work of the Modern Science Club, of Brooklyn, are the visits to the various power plants in that city. On the evening of February 24, after dining at the club house, the members visited the Bay Ridge plant of the Brooklyn Edison Company, and a thorough inspection of the plant was made under the guidance of Charles H. Courser, assistant engineer of the company, and George L. Horton, chief engineer of the plant.

The Iowa Engineering Society met at Cedar Rapids, February 16 and 17. Several papers of a civil-engineering character were read at the meeting. The two-day session wound up with a trolley excursion to Iowa City, where the final exercises took place in the Physics building of the University of Iowa. Officers for the ensuing year were elected as follows: T. R. Warriner, Cedar Rapids, Iowa, president; Seth Dean, Glenwood, Iowa, vice-president; directors, S. N. Williams, Mt. Vernon, and L. B. Spinney, of Ames; secretary, S. M. Woodward, Iowa City.

The regular monthly meeting of the Engineers' Society of Western Pennsylvania was held in the society rooms, Fulton building, Pittsburg, February 15. Elmer K. Hiles was reëlected secretary for the ensuing year. A paper on "The Government Testing Station in Pittsburg" was presented by J. C. Roberts, assistant chief engineer, Technological Branch, United States Geological Survey. Lantern slides were shown illustrating apparatus in use at the station, and a very illuminating description of the work accomplished as well as that to be taken up in connection with the testing of structural materials, fuels and the causes and prevention of mine accidents was given.

The smoker of the Park Row Branch of The Technical League, was held at the Teutonia Assembly rooms, 158 Third avenue, New York City, on Saturday evening, February 19. Fully 250 members and friends gathered together, and a most pleasant night was spent. There were recitations, stories, monologues, musical selections, and songs in which all joined in the chorus. During the evening, Walter G. Eliot, a prominent member, spoke earnestly on the aims and objects of the society; and told his hearers that besides many other important features, the first great effort of the League should be to have laws enacted in the various States, making engineering a closed profession, whereby no one may practice engineering without a license.

On February 23, Melville Council, No. 9, U. C. C. of E., held its first annual ladies night at the new meeting rooms of the association, 315 Washington street, Brooklyn, N. Y. The spacious rooms were comfortably filled. The "bunch" furnished the entertainment and were assisted by local talent, friends of the council, including master Waldorf Schabacker, recitation; Miss Lent, piano solo; Miss Schabacker, recitation; Miss Ethel Martin and Mabel Brush, piano duet; Misses Martha Rosenberg and Florence Olsen, piano duet. Past-president Stone, of the N. A. S. E., addressed the gathering, after which refreshments were served. The committee in charge were Thomas H. Lomas, chairman; George H. Richdale, Charles Isakson and George O. Kaley.

One of the most ambitious jaunts ever undertaken by the Chicago N. A. S. E. visiting club, occurred Saturday evening, February 26, when the club visited Illinois No. 10, of Ottawa, 80 miles west of Chicago. Mayor Farrell, of Ottawa, welcomed the visitors and Representative Brown, of the State legislature, was present at the meeting. Mr. Brown became much interested in the State license question, discussing it at length, offering many valuable suggestions in regard to the manner of presenting the bill and pledging his support at the proper time. It was considered by all that the visit was a great success, inasmuch as it will result in favorable influence being brought to bear on the legisative questions which engineers desire so much to see passed in the State. Twenty-two boosters made the trip out from Chicago.

The fifteenth annual banquet of Phoenix Association, No. 24, N. A. S. E., of New York City, was held at the Broadway Central hotel, Thursday evening, February 24. More than four hundred were seated at the tables to enjoy the dinner, among them being delegations from Philadelphia, Boston and other nearby cities. There were also in attendance W. J. Reynolds, national president, and past national presidents, Joseph F. Carney, Herbert, E. Stone and William T. Wheeler, and other national officers. Instrumental and vocal musical numbers enlivened the occasion. Walter M. Brainerd, president of the local order, acted as toastmaster, and read congratulatory telegrams from President Taft, Governor Hughes, the Chicago Engineers, Pennsylvania Association, No. 16, National Secretary Fred Raven and John W. Lane, editor of the *National Engineer*.

The second annual entertainment and reception of the combined associations of engineers of the borough of Brooklyn, N. Y., was held at the Fourteenth Regiment armory on Monday evening, February 21. The combination comprises the Modern Science Club, Brooklyn Council No. 8 and Melville Council No. 9 of the Universal Craftsmen, Nos. 8, 27, 41 and 57 of the National Association of Stationary Engineers and the Associated Board of Engineers. Company D, of the Fourteenth Regiment, was also on this occasion included in the combination. The large drill room was tastefully decorated especially for this event. About three thousand members and friends were present, including many of the most prominent personages in N. A. S. E. circles, the nearby cities being largely represented. A high-class vaudeville performance was followed by dancing. The several committees deserved the congratulations bestowed upon them for their earnest work in making the occasion a success. The officers are: William Smith, chairman; George W. Roff, vice-chairman; William F. Brundage, secretary; Charles Schabacker, financial secretary; George O. Kaley, treasurer.

A resident of New Tazewell, Tenn., claims to have solved the problem of perpetual motion and believes that he will have a perfected machine ready in a short time. He claims that the discovery is due to divine revelation, and that the discovery is not to be brought about for any man's personal gain, but for the good of humanity. At any rate his intentions are good.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering building, 29 West 39th St., New York. Monthly meetings in New York City. Spring meeting at Atlantic City, May 31 to June 6.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff, Denver, Colo.; sec. and treas., Frank M. Tait. Association York. Next annual convention, St. Louis, Mo., May 23-28.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, 803 Fulton building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., William G. Snow; sec., William M. Mackay, P. O. Box 1818, New York City.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, W. S. Cadwell, Chicago, Ill.; sec., Thomas H. Jones, 244 Eighth street, N. E., Washington, D. C. Next convention, Buffalo, N. Y., August, 1910.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 525 Manhattan building, Chicago, Ill. Next convention, Rochester, N. Y., September, 1910.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next convention, Philadelphia, Pa., June, 1910.

OHIO SOCIETY OF MECHANICAL, ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. E. Brown; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Niagara Falls, Canada, May 24-27, 1910.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee, Peoria, Ill. Next convention, Denver, Colo. September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., A. C. Rogers, Toledo, O.; sec. and treas., D. L. Gaskill, Greenville, O. Next annual meeting at Toledo, O., May, 1910.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

STEAM, WATER AND GAS PRIME MOVERS

INTERNAL-COMBUSTION ENGINE. Henry A. Burt, Detroit, Mich. 948,308.

INTERNAL-COMBUSTION ENGINE. Edward Moore, Syracuse, N. Y. 948,336.

WATER MOTOR. Lee L. Dodds, Columbus, Ohio, assignor to the Monarch Specialty Manufacturing Company, Columbus, Ohio, a Corporation of Ohio. 948,501.

ROTARY ENGINE. Willis G. Dodd, San Francisco, Cal. 948,692.

WATER WHEEL. Fred Bangerter, New York, N. Y., assignor to the Standard Water Meter Company, Brooklyn, N. Y., a Corporation of New York. 948,755.

TWO-STROKE CYCLE GAS ENGINE. John M. Johnson, Alfred C. Johnson and August Johnson, Sunnyvale, Cal. 948,860.

TURBINE. George A. Boddy, River Rouge, Mich., assignor of one-fourth to John B. Kessler, one-fourth to George H. Boddy, and one-fourth to Ernest R. Boddy, River Rouge, Mich. 948,916.

EXPLOSIVE ENGINE. Herbert G. Holmes, Lansing, Mich., assignor to Ideal Motor Company, Lansing, Mich., a Corporation. 948,925.

GAS ENGINE. Dwight H. Bennett, Brooklyn, N. Y. 948,950.

BOILERS

TUBULAR STEAM BOILER. Sterling Elliott, Newton, Mass. 948,373.

PIPING, VALVES AND TOOLS

WRENCH. Fred Knobel, Odessa, Neb. 948,332.

GATE VALVE. Alexander Watson, Beachmont, Mass., assignor of one-third to Byron C. Leavitt, Duxbury, Mass., and onet-hird to William C. Mitchell, Malden, Mass. 948,359.

MACHINE FOR THREADING PIPES. Emil A. Hoeffer, Freeport, Ill., assignor to the Borden Company, Warren, Ohio, a Corporation of Ohio. 948,394.

HOSE COUPLING. Clarence F. Wilson and John Oldenburg, Johnstown, Colo. 948,437.

NONRETURN VALVE MECHANISM. Chas. E. Golden, Crafton, Penn. 948,445.

VALVE. George B. Kennington, Pittsburg, Penn., assignor to Golden-Anderson Valve Specialty Company, Pittsburg, Penn., a Corporation of Pennsylvania. 948,452.

PNEUMATIC SAFETY VALVE. Frank Moyle, Jamestown, Cal. 948,469.

UNIVERSAL CONTROLLING VALVE. William C. Arp, Fred F. Hildreth, and Frederick N. Rumbley, Terre Haute, Ind. 948,571.

ROD PACKING. Thos. A. Johnston, Chadron, and Thomas L. Finley, Long Pine, Neb., assignors of three-fourths to said Johnston and one-fourth to said Finley. 948,610.

VALVE. Carl W. A. Koelkebeck, Cleveland, Ohio. 948,649.

HOSE COUPLING. Egbert H. Gold, Chicago, Ill. 948,667.

BOILER CHECK VALVE, CLEAN-OUT. John R. Martin, Clarksville, Tenn. 948,702.

THREAD-CUTTING TOOL. John B. Phillips, Warren, Ohio, assignor to the Borden Company, Warren, Ohio, a Corporation of Ohio. 948,706.

SOOT EJECTOR FOR WATER-TUBE BOILERS. John J. Hanlon, Troy, N. Y. 948,761.

NIPPLE-THREADING MECHANISM. Herman W. Oster, Cleveland, Ohio, assignor to the Oster Manufacturing Company, Cleveland, Ohio, a Corporation of Ohio. 948,793.

WRENCH. Livy L. Barrick, Moundsville, W. Va., assignor of one-third to Clarence G. Dawson, Moundsville, W. Va., and one-third to Adolph Loehl, Washington, D. C. 948,870.

VALVE GEAR. John Irving, Cardross, Scotland. 948,976.

ELECTRICAL DEVICES AND APPLIANCES

CARBON-FEEDING MECHANISM FOR ARC-LAMPS. Ladislav O. Kozar, Wilkes-Barre, Penn., assignor of one-third to Joseph Murgas, Wilkes-Barre, Penn. 945,960.

ELECTRICAL APPARATUS FOR MELTING SEALING-WAX. William T. Von Tillow, San Francisco, Cal. 945,822.

MECHANICAL CONSTRUCTION FOR STORAGE-BATTERY CELLS. Patrick Kennedy, New York, N. Y., assignor to Consolidated Railway Electric Lighting and Equipment Company, New York, N. Y., a Corporation of Maine. 945,397.

ELECTRIC SWITCH. William T. Arndt, Shreveport, La. 945,011.

ELECTRIC TIME-SWITCH MECHANISM. William C. Blundell, Kansas City, Mo. 945,013.

FUSE-BOX. Robert C. Cole, West Hartford, Conn., assignor to the Johns-Pratt Company, Hartford, Conn. 945,017.

ELECTRIC FURNACE AND METHOD OF OPERATING THE SAME. Augustin L. J. Queneau, Philadelphia, Penn. 948,343.

ELECTRIC-MOTOR-CONTROLLING DEVICE. Frank A. Hayes, New York, N. Y. 948,375.

BATTERY ELECTRODE SUPPORT. Charles B. Schoenmehl and Wilfred Straw, Waterbury, Conn.; said Straw assignor to said Schoenmehl. 948,379.

ELECTRIC SWITCH. Daniel G. Butts, Chicago, Ill., assignor to General Electric Company, a Corporation of New York. 948,441.

STORAGE-BATTERY ELECTRODE. Thomas A. Edison, Llewellyn Park, Orange, N. J., assignor to Edison Storage Battery Company, West Orange, N. J., a Corporation of New Jersey. 948,558.

AUTOMATIC MOTOR CONTROLLER. Arthur E. Handy, Cranston, R. I. 948,560.

APPARATUS FOR ELECTROSTATIC-MAGNETIC SEPARATION. Henry M. Sutton, Walter L. Steele, and Edwin G. Steele, Dallas, Tex. 948,599.

ELECTROMAGNETIC SWITCH. Roy V. Collins, New York, N. Y., assignor to American Automatic Switch Company, New York, N. Y., a Corporation of New York. 948,624.

ACCUMULATOR. Max Schneider, Radebeul, near Dresden, Germany. 948,639.

ELECTRIC-LAMP SOCKET. William S. McLewee, Yardley, Penn., assignor to Carrie Howell McLewee, Yardley, Penn. 948,670.

CIRCUIT BREAKER FOR ELECTRICALLY HEATED DEVICES. William G. Nagel, Toledo, Ohio. 948,704.

INDUCTOR GENERATOR FOR IGNITION PURPOSES. Henry J. Podlesak, Chicago, Ill., and Tesla E. Podlesak, Morristown, N. J. 948,483.

INSULATING MEANS FOR FLUSH SWITCHES. Clarence D. Platt, Bridgeport, Conn. 948,707.

SYSTEM OF ELECTRIC GENERATION, DISTRIBUTION AND CONTROL. Lamar Lyndon, East Orange, N. J. 948,736.

INCANDESCENT-LAMP SOCKET. Fredrick C. Aschenbrener, Chicago, Ill., assignor of one-fourth to himself, one-fourth to Oscar H. E. Fritz, one-half to Joseph C. Seyl, Chicago, Ill. 948,754.

DYNAMO ELECTRIC GENERATOR. Geo. Lewis, Newark, N. J. 948,817.

CONDUIT FOR ELECTRIC CONDUCTORS. George A. Lutz, Plainfield, N. J., assignor to American Circular Loom Company, Portland, Maine, a Corporation of Maine. 948,865.

ARC-LIGHT ELECTRODE. Isador Ladoff, Cleveland, Ohio, assignor of thirty one-hundreths to Philip C. Peck, New York, N. Y. 948,296.

ARC LAMP. Mariano Fortuny, Paris, France. 948,971.

MISCELLANEOUS

ALCOHOL GAS BURNER. Wesley E. Cochran, Montgomery, Ala., assignor to the Howler Manufacturing Company, Montgomery, Ala. 948,982.

ARTIFICIAL FUEL AND PROCESS FOR MAKING THE SAME. Charles F. Bonhack, New York, N. Y. 948,812.

CARBURETER. William P. Kingsbury, Port Arthur, Tex. 948,977.

CARBURETER FOR COMBUSTION ENGINES. Richard E. Krause, Cleveland, Ohio, assignor to the Krause Carbureter Company, Cleveland, Ohio, a Corporation of Ohio. 948,612.

CYLINDER LUBRICATOR. Charles C. League, Renovo, Penn. 948,455.

CYLINDER LUBRICATOR. Frank B. Hubbard, Middlefield, Ohio. 948,882.

GUARD FOR WATER GAGES. David Buissono, Blue Island, Ill. 948,412.

JOURNAL BRASS. Frank J. Finning, Nashua, N. H. 948,316.

OIL BURNER. William H. Creswell, Los Angeles, Cal. 948,921.

PACKING RING. Louis E. McQuay and William E. Norris, East St. Louis, Ill. 948,518.

VAPORIZING AND MIXING DEVICE. Charles F. Preston, La Grange, Ill. 948,402.

MOMENTS WITH THE Advertising Editor

A Department for Subscribers Conducted by the Business End of Power and the Engineer

Speaking of advertising,—said Bill,—always reminds me of the story of my friend, Jim, who's engineer over at the Smithtown Company works now. Jim comes in here one day and says to me, "Bill, I'm in trouble."

"What's the matter?" says I, "Wife got the tantrums again or kid got the colic?" It always seems to me that either women or kids are at the bottom of all trouble.

"Nope," answers Jim, "It's just this way. I can't seem to keep things right in my plant.

If the boiler's getting along all right, the pipes start a leak; get the pipes tight and the engine begins to groan; get her lubricated and the boiler feed pump goes on strike. Oh, it's a swell job being a stationary engineer, all right, all right."

"The job's all right, my man," says I. "It's the engineer that needs fixing. Now, why in the name of Christmas don't you get things ship-shape? Why don't you get the right stuff to fix these troubles and end 'em."

"That's easy enough to say," grunts Jim, "but it's the doing it that forms the ice over my mental process."

"It's easier than beating the beer-drinking record on a hot day," I answers. "Here's what you do. Get your last copy of POWER. Read the ads. Study them. See what the different manufacturers claim. Sift the claims. Then apply the result to your troubles. After you've studied the matter take it up with the Old Man and let him see that it's to his interest to furnish his plant with modern equipment."

Jim gives me the kind of look the heroine gives the villain when he asks her to stroll with him in the shady wood. "Me read the ads., Jim!" he cries. "Me! I ain't got any use at all for advertising." "And why not," I ask, too surprised to argue right then.

"For a very good reason," he answers. "Ten years ago my wife answered an ad. of some investment company and after several letters to and fro sent a hundred dollars of my savings unknown to me. Well, that was the last we ever heard of the hundred. And I ain't had any use for advertising since."

"Jim," I says, speaking slow so it would sink in, "I think you're the Original Grouch. Don't you know you can't condemn the whole of advertising on account of one fake ad., just the same as you can't condemn all insurance just because of one bum policy." "That's all right," says Jim, "I've paid for my lesson, but I've learned it, and nothing you can say, Bill, will change me."

"There's another thing," I went on, getting a little hot under the collar, "advertising has gotten about thirty years honester in the last ten years. Not because the advertisers got a sudden rush of virtue to the pen, but because they found honest advertising was the only kind that paid, the nearer it stuck to the truth, the *better* it paid. Now, you go right back to your job and read the ads. in POWER, bearing in mind what I told you."

Jim shook his head. "I know you mean well, Bill," he says, "but it's a fool dog that bites of poisoned meat twice." "Yes, and it's a damn fool dog that refuses to eat ever after on that account," says I. Jim gets huffed at that and flings out the door without even saying good-day.

Well, things came up after that keeping me pretty busy, and I didn't see Jim again for six months or more. Finally, I got off one afternoon and goes across town to the Smithtown Company plant to see him. When I got in the engine-room, there he was, smoking a pipe, his chair tilted back against the wall, and reading the advertising pages of POWER.

"Well, Jim," I ask, "How's the world treating you now?"

"Fine! Fine!" says Jim.

"Having any trouble with your engine or boiler?" I ask.

"No, I've been making a study of these advertised devices in POWER lately," he says. "I've suggested some to the Old Man and he's let me put in quite a few and throw out the old ones. He's saving money, and what's more, knows I've saved it for him, and I'm saved trouble. And everybody's happy!" "But," says I, feeling as if somebody had landed me a blow in the solar, "I thought you didn't believe in advertising!" "Me!" says Jim, looking as if I'd told him he didn't believe in steam, "You're thinking of some other man, Bill!" No, I don't believe he was lying—concluded Bill, after a long pause—I just think he was so plumb grateful to me he was ashamed to admit it.